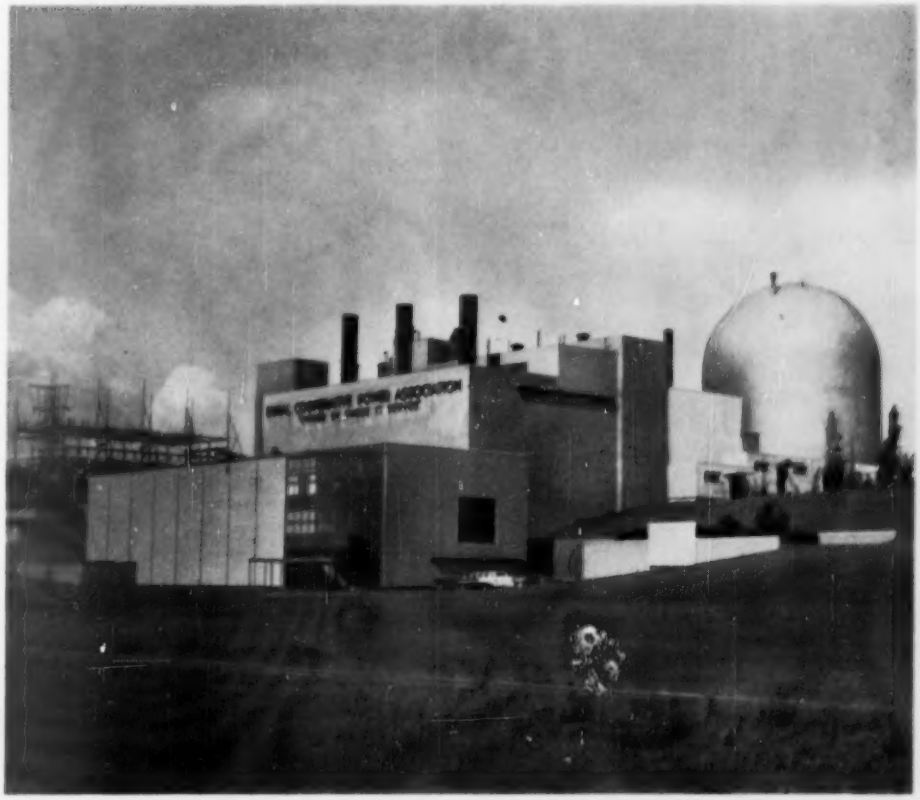
Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION



May 1961

Experiences Dictate Precipitator Improvements

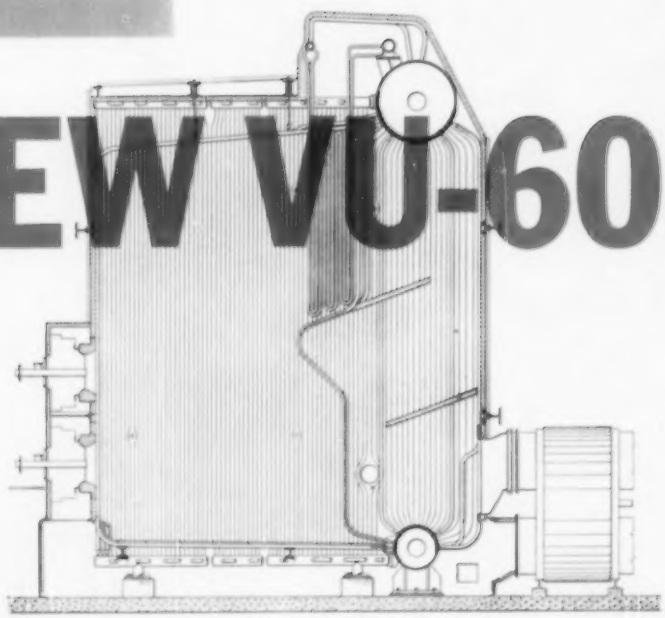
Economic Insulation Thickness

Fluid Drives For Turbine Generator Driven Pumps

American Power Conference in Review—II

Now...a completely self-cased, MODULAR BOILER

C-E's NEW VU-60



DESIGN HIGHLIGHTS OF THE VU-60

Modular construction makes it possible to assemble custom designs from standardized components.

Boiler can be proportioned for best combustion, heat absorption and gas flow even when space conditions are limited or difficult.

Completely self-cased, the VU-60 is an all-welded gas-tight and pressure-tight envelope of finned furnace and boiler tubes. This results in less non-working weight per pound of steam capacity.

No external ductwork. The bottom of the boiler is a plenum chamber which allows air to flow direct from the air heater to the burner windbox.

Number of field-welded pressure parts is greatly reduced.

Cross flow baffling and symmetrical boiler bank mean low draft loss, no sluggish gas flow in any pass.

Requires only a simple, reinforced concrete slab. No conventional roller supports required.

Paneled construction cuts erection time.

No air heater supporting steel is required.

For pressurized or induced draft firing.

VU-60 SPECIFICATIONS

Capacities:	100,000 to 250,000 lb per hr
Design pressures:	250, 500, 750, 1000 psi
Steam temperatures:	To 900 F
Fuels:	Oil and/or gas
Firing:	Horizontal (front wall) or tangential
Size increments:	Depth – twelve Width – eight Height – three
Stream drum sizes:	Four

The VU-60 Boiler is a new and logical extension of C-E's service-proved VU series. Like all VU units, it is completely integrated, symmetrical in design, uncluttered and uncomplicated. Based on a standardized modular concept, it offers greater flexibility, yet stays within the bounds of solid and proven practice.

The VU-60 is a functional boiler. Completely self-cased, the exterior walls of this unit (furnace and boiler) are composed of finned tubes, shop welded into panels. The fins between panels are field welded to form a gas- and pressure-tight structure. Insulation is applied directly to the outside of these walls and is covered by pre-formed, field installed metal lagging. Thus non-working weight and bulk is reduced to a minimum.

The VU-60 is economical, it is dependable, it is accessible. It is easy to install and will meet your most exacting steam needs. Why not write for further information, or, if you prefer, contact the C-E office nearest you.

COMBUSTION ENGINEERING

C-E

General Offices: Windsor, Conn.
New York Offices: 200 Madison Avenue, New York 16, N. Y.
Canada: Combustion Engineering-Superheater Ltd.

C-300A

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

volume 32 number 11 **May 1961**

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BPA

Printed in U. S. A.

COVER PHOTO

Construction is reported almost complete at Elk River, Minnesota, rural electric atomic plant

(Photo courtesy Allis-Chalmers Mfg. Corp.)

Observations and Experiences Resulting From A Precipitator Improvement Program . . . 22

M. J. Archbold

There have been many proposals, and fewer attempts, to establish some standards for precipitator performance. This paper, a follow-up to an earlier, basic offering by the same author, gives performance data as against equipment criteria to permit intelligent comparisons.

Economic Thickness of Insulation For Flat Plates and Pipes . . . 33

W. C. Turner

As the author states: "Depending upon cost of insulation and the cost of heat, a certain definite thickness of insulation provides the lowest total cost." He then describes the procedure to determine this thickness.

Fluid Drives for Turbine-Generator Driven Boiler Feed Pumps . . . 38

Richard D. O'Neil

On-the-shaft drives whether mounted on the turbine-end or the generator-end for boiler feed pumps will make their mark for reliability as much as for economy. Here is a brief description of most of the early applications.

American Power Conference In Review . . . 46

Abstracts From the Technical Press—At Home and Abroad . . . 55

Editorial: Air Pollution Control—A Joint Responsibility? . . . 21

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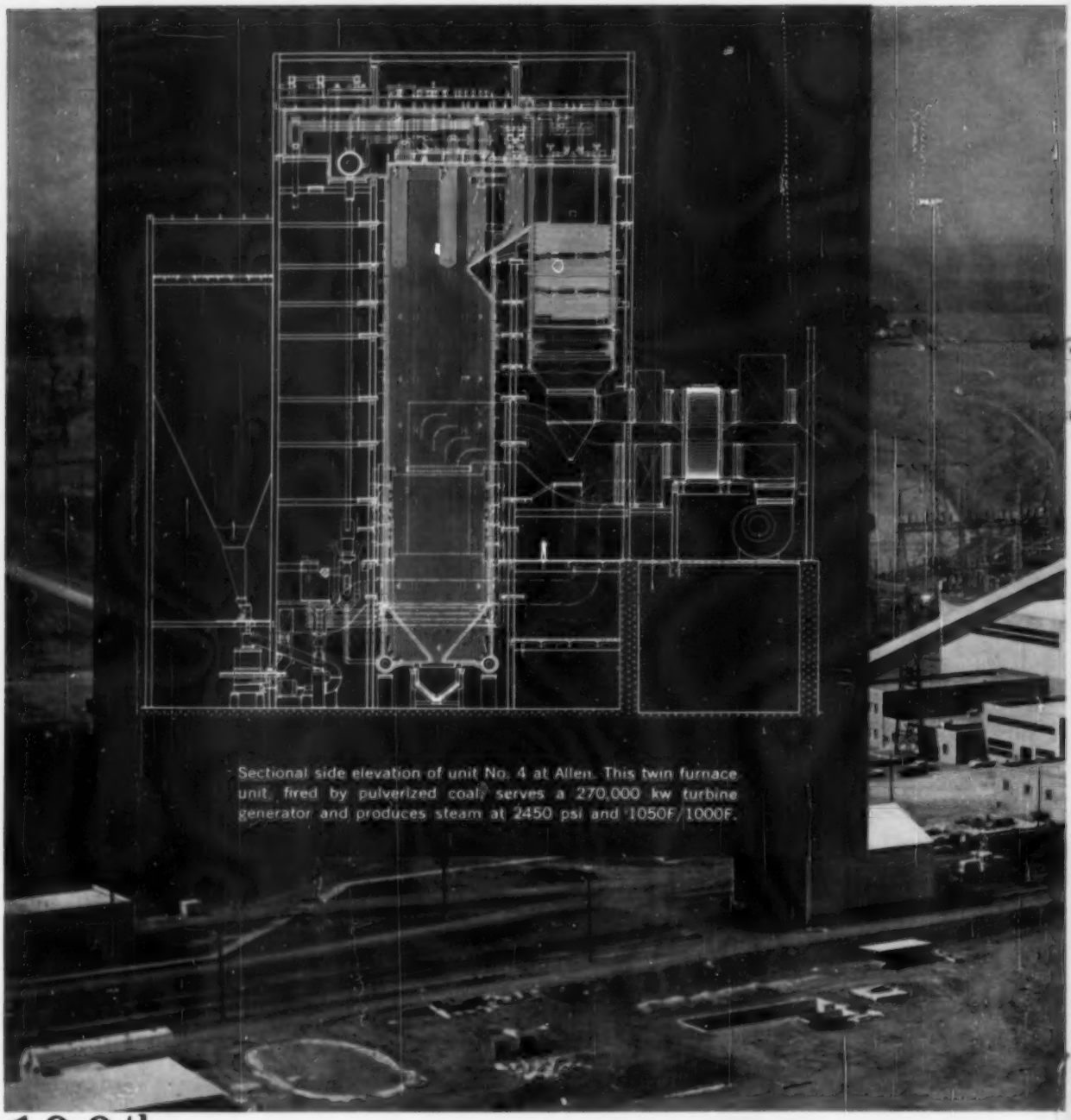
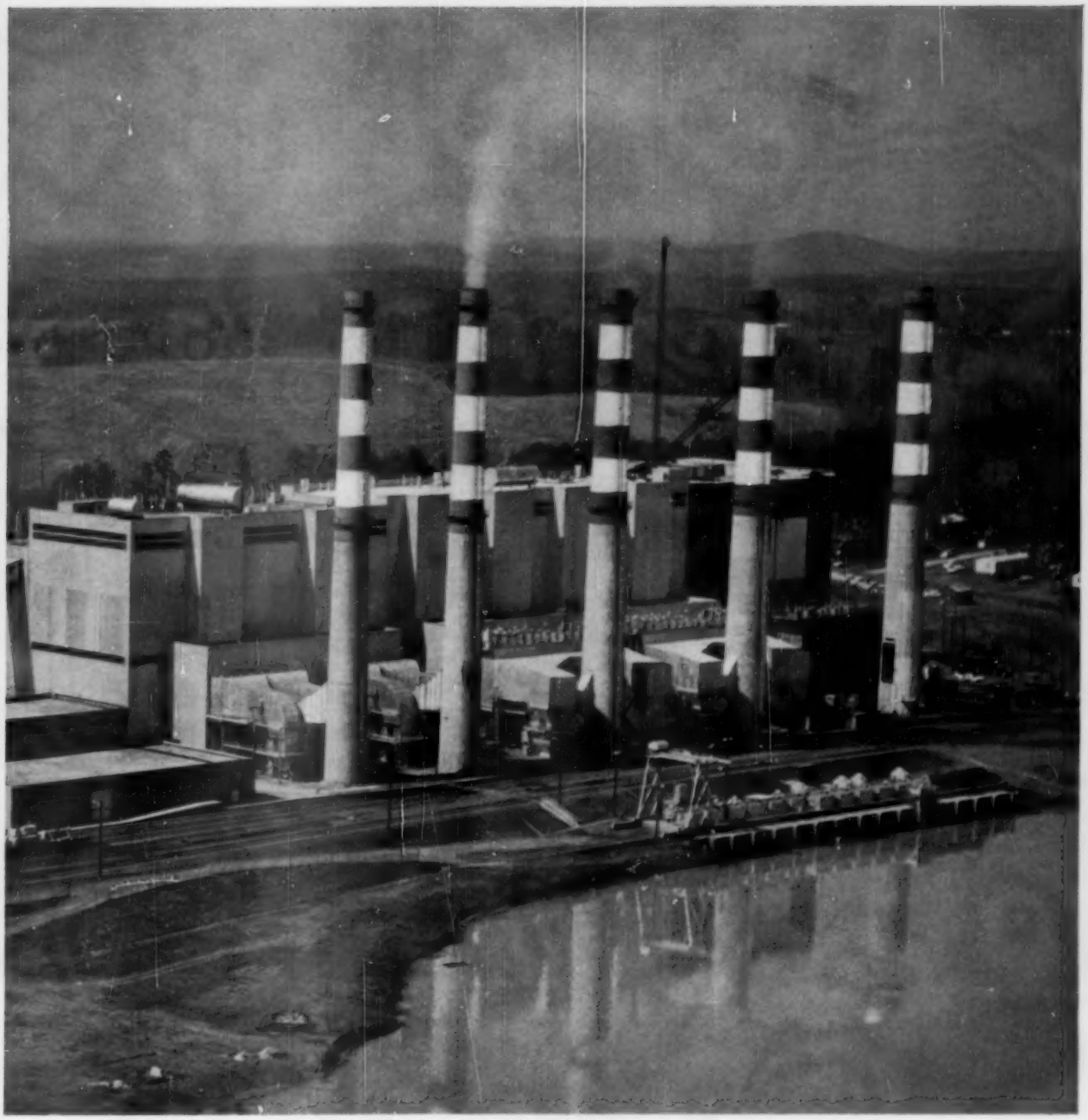


Sectional side elevation of unit No. 4 at Allen. This twin furnace unit, fired by pulverized coal, serves a 270,000 kw turbine generator and produces steam at 2450 psi and 1050F/1000F.

100th C-E CONTROLLED CIRCULATION

Late in 1960, C-E's one hundredth Controlled Circulation Steam Generator was placed in operation. Located at Duke Power Company's Allen Plant, this, the fourth Controlled Circulation unit in service at Allen, will soon be joined by a fifth, now under construction. The Duke Power system presently has more than 2,300 mw of Controlled Circulation generating capacity, installed or on order.

Since the first postwar C-E Controlled Circulation unit was placed in service about eight years ago, this type of unit has established itself as the design best suited for the 2400 lb throttle pressure



UNIT NOW IN SERVICE

range, and has achieved unparalleled acceptance throughout the utility industry. Today, units of this design are on order, under construction, or installed in utility plants the world over. The total capacity of the turbine generators served is about 30,000 mw.

COMBUSTION ENGINEERING

C-314

General offices: Windsor, Connecticut
New York offices: 200 Madison Avenue, New York 16

PAPER MILL EQUIPMENT; PULVERIZERS; FLASH DRYING SYSTEMS; PRESSURE VESSELS; SOIL PIPE

IT'S NOT TOO EARLY TO THINK OF NEXT WINTER'S COAL HANDLING PROBLEMS

PENNSYLVANIA FROZEN COAL CRACKERS WILL PREVENT

- **Hopper jams**
- **Overloaded conveyors**
- **Slowdowns**
- **High labor costs**

Get ready *now*—for the worst conditions next winter can bring. Investigate Pennsylvania Frozen Coal Crackers.

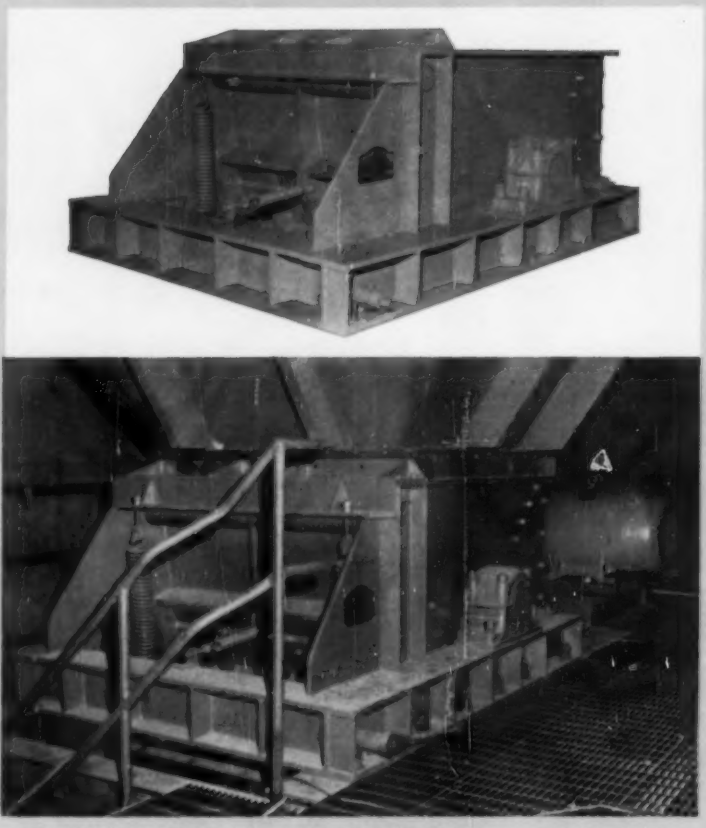
These are the crushers especially designed and widely used for handling frozen coal at track delivery points and storage yards.

OPERATION

Wide feed openings permit use of *open-throated hoppers.* Frozen masses of coal are dumped (with rotary car dumpers) directly into hoppers. No need for grates. Frozen coal works its way freely to the hopper bottom where the crackers' special teeth break it up.

Result—uniform product with limited top-size and minimum fines is fed conveyors, providing protection from overloads . . . insuring continuous flow of coal.

Breaker plates can be opened in warm weather, permitting coal to pass without operating crusher.



30 x 60 Single Roll Pennsylvania Frozen Coal Cracker in service at Merrimack Steam Station, Public Service Company of New Hampshire, Concord, N.H.

OTHER FEATURES

Built-in tramp iron protection. Low head room requirements. Easily adapted to existing hoppers. Capacities up to 1200 TPH. Available in single or twin units.

Write today for new Pennsylvania Bulletin 2013 . . . or call your nearest Pennsylvania representative.

PENNSYLVANIA CRUSHER DIVISION

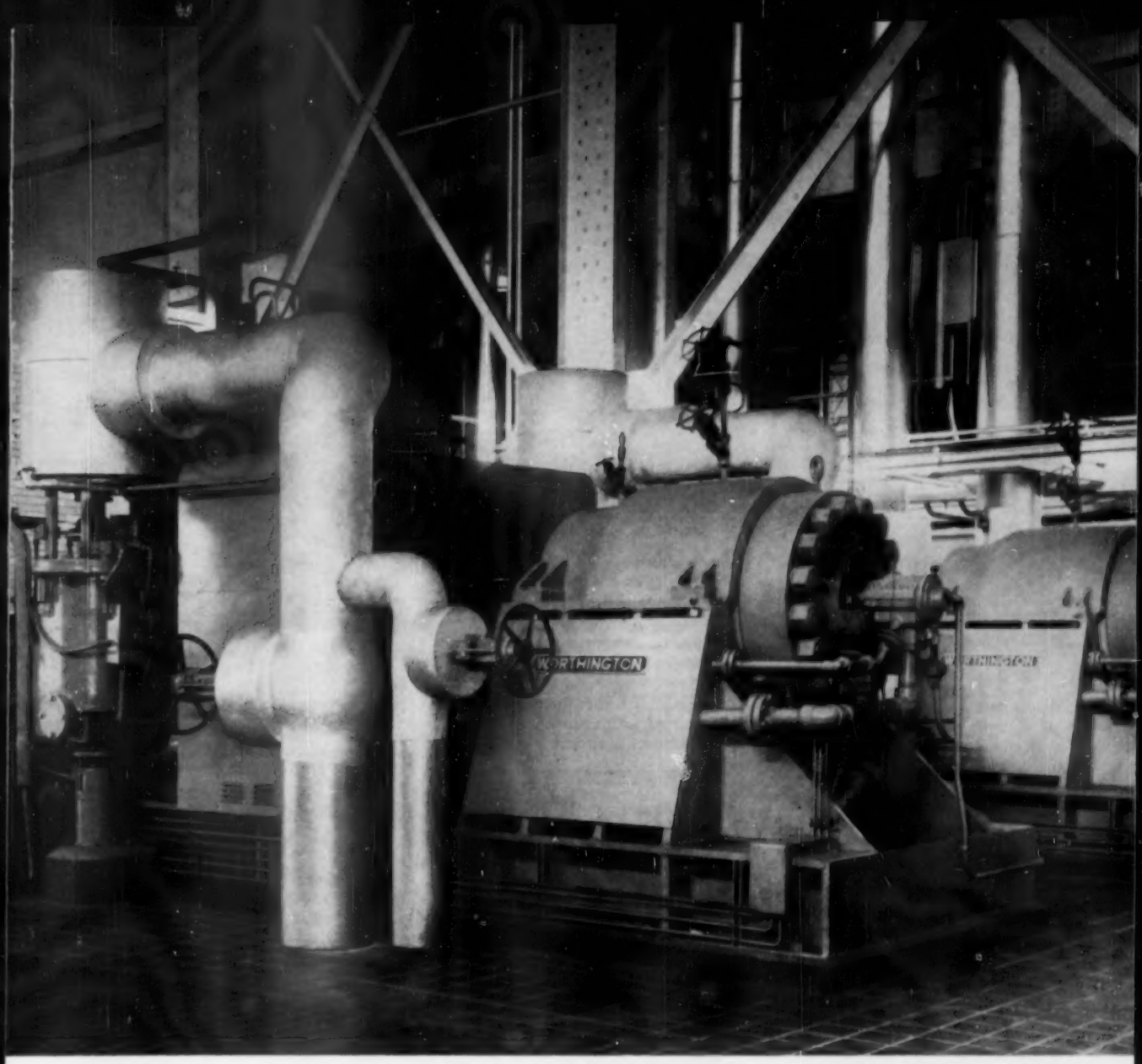
BATH IRON WORKS CORPORATION
WEST CHESTER, PENNA.

★ ★ ★

Over 50 years concentrated experience in mechanical reduction. Call on Pennsylvania with your next crushing problem. Representatives from coast-to-coast.

PENNSYLVANIA CRUSHERS

BATH-BUILT



NEW PUMPS SET WORLD RECORD

Three Worthington high-speed boiler feed pumps have set a new world pumping record—developing a net pressure of 4514 psi in six stages within one casing. They constitute the first pump installation to develop the entire pressure required to feed a supercritical steam generator in one step.

They are located at Avon Unit No. 8, a 250,000 kw supercritical pressure steam generating unit of the Avon Plant of Cleveland Electrical Illuminating Company. The maximum turbine flow is 1,715,000 lbs. of steam per hour at 3500 psig.

The three pumps deliver 2285 gpm each at a total head of 11,580 feet. Two pumps are driven by steam turbines. The standby and start-up pump is motor-driven through a hydraulic coupling and a step-up gear.

In earlier supercritical pressure steam power plants, two or more boiler feed pumps operating in series have been used to build the total boiler feed pressure. But Cleveland Electric's studies disclosed that the higher cost of closed heaters designed for the full discharge pressure was compensated by the savings in pump and driver costs and the power savings of a one-step pumping system, such as they finally selected.

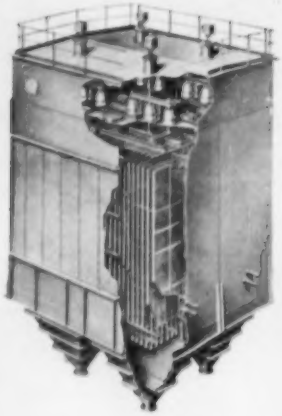
Should you select a one-step pumping system or a series system for your high-speed supercritical steam plant? Ask your Worthington representative for a complete installation report on the Avon Station Unit No. 8. And they will be happy to supply further information for your own studies. Or write Worthington Corporation, Harrison, New Jersey. In Canada, Worthington (Canada), Ltd., Brantford, Ontario.

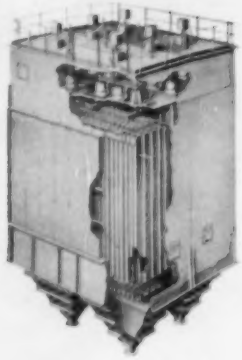
WORTHINGTON

PRODUCTS THAT WORK FOR YOUR PROFIT

How Important Is Precipitator Size?

Traditional Design →





← Modern Design

As shown in these two illustrations, Research-Cottrell's more compact modern design is appreciably smaller than the traditional design.

A common fallacy assumes a larger precipitator with lower gas velocity is preferable to a smaller precipitator with higher gas velocity.

This assumption overlooks important economies now obtained with:

(a) Efficient voltage application.

(b) Specific control of electrical sections.

(c) Electrode design improvements.

(d) Automatic Rapper Control.

(e) Accurate internal baffling.

(f) Proper control of gas flow, based on three-dimensional model studies.

Close familiarity with these techniques and improvements has made possible guarantees of 99 per cent collection on large, modern installations.

A. Applied Voltage

The relationship between voltage and size of the precipitator is approximately as shown in Figure 1.

Most efficient operation and minimum size are obtainable by insuring operation at maximum voltage under varying conditions of gas flow, particle size, resistivity, etc. Generally, the amount of arc-over in the precipitator will determine the practical operating voltage limit. By using the arc-over as the motivating control factor, through measurement and integration, maximum voltage can be insured at all times.

B. Sectional Voltage Control

Since gas conditions vary throughout any precipitator, higher efficiencies are obtainable if the precipitator is sectionalized with individual voltage control for each section.

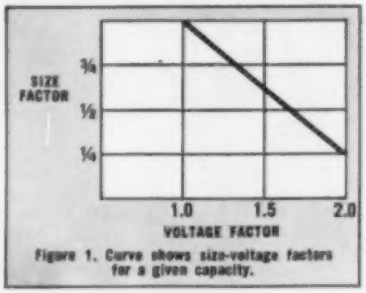


Figure 1. Curve shows size-voltage factors for a given capacity.

C. Electrode Design

Efficiency can be seriously affected by re-entrainment (that is the re-suspension of dust that has already been collected). Properly designed collecting electrodes, or plates, providing correct baffling to shield the collecting surface against the sweeping and scouring effects of the gas stream can reduce overall precipitator size 25–50 per cent.

D. Automatic Rapper Control

Rapping intensity must vary to meet the changing conditions of gas flow, particle size and resistivity. Otherwise, efficiency will decrease due to either too heavy a blow causing a re-entrainment, or too light a blow allowing dust build up on the electrodes necessitating operation at reduced voltages. Automatically controlled, fully adjustable rapping equipment insures optimum operation at all times.

E. Internal Baffling

Maximum efficiency can only be obtained by reducing the sneakage (part of gas stream which leaks or sneaks around the treatment zone) to an absolute minimum through the proper design and use of internal baffling proven by model studies and research on full size precipitators.

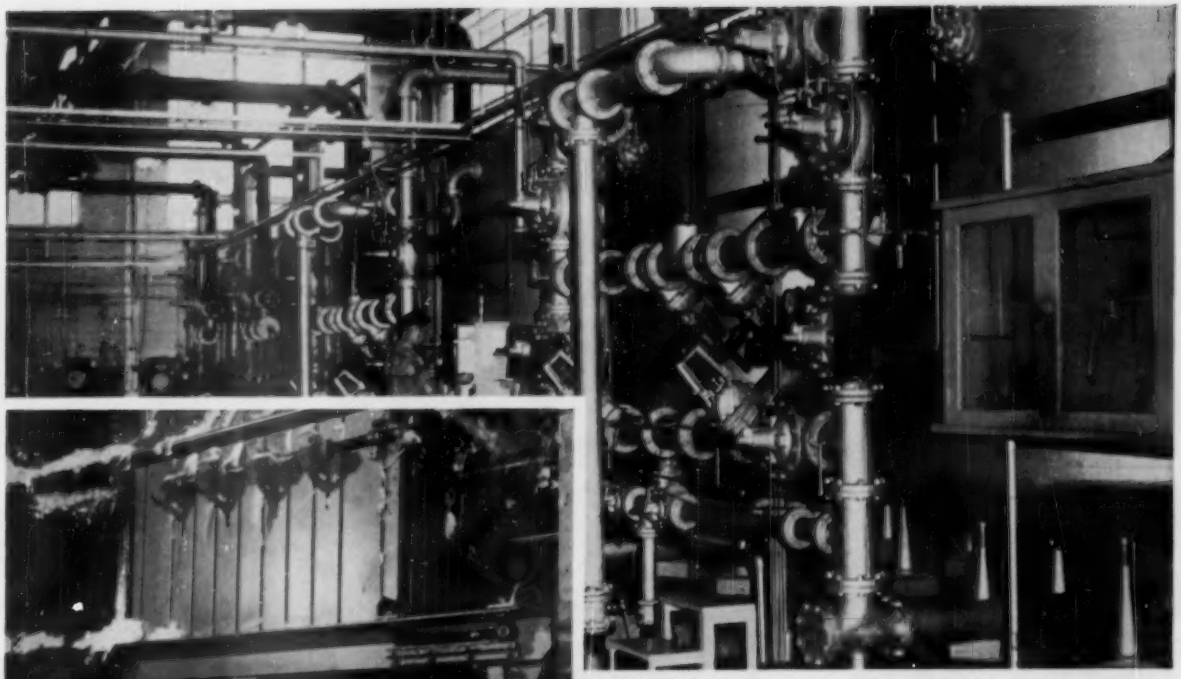
F. Gas Flow—Model Studies

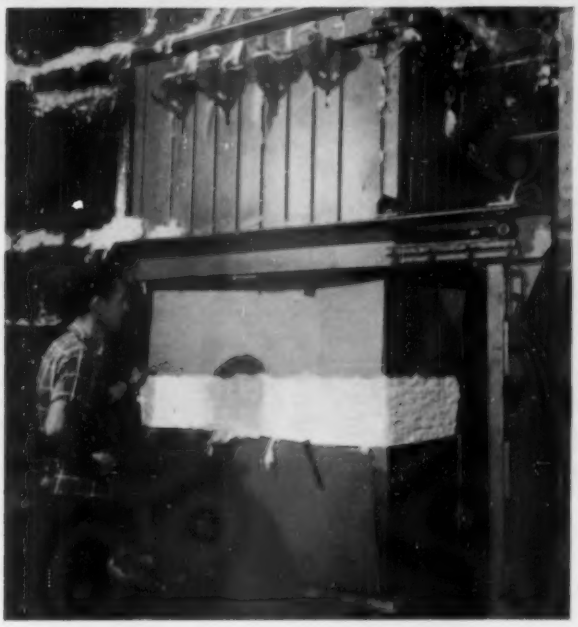
Proper gas flow patterns across the face of the precipitator can result in a 25–50 per cent reduction in over-all size. The use of transparent plastic three-dimensional model studies as well as field research has resulted in refinements in design never before possible.

Only at Research-Cottrell can you be assured of these latest of design techniques and improvements providing the most economical selection of equipment and efficiency of operation. Research-Cottrell personnel, backed by nearly a half century of gas cleaning experience, stand ready at any time to provide information or service.

For details or more information relative to the data contained herein, or any other general inquiries, please contact your nearest Research-Cottrell representative or write directly to:

Research-Cottrell, Inc.
Office and Plant: Bound Brook, N. J.
Representatives in Principal Cities of U.S. and Canada. RC233





ABOVE: AMBERLITE® 200 is used in two softeners in this group. Here the durable polymeric structure of this resin successfully resists the action of "aggressive water" which shortens useful life of conventional ion exchange resins.

LEFT: Baling of rayon staple fiber at Lowland, Tenn., plant of American Enka Corporation.

Enka rayon plant solves "Aggressive Water" problem with AMBERLITE 200

At the Lowland, Tenn., plant of American Enka Corporation, water for washing rayon staple fiber and yarn is breakpoint chlorinated to oxidize impurities. It enters softeners with 0.3 ppm residual chlorine. Trace metals present in the raw water act as oxidation catalysts and make the chlorine "aggressive" toward ion exchange resins. Useful life of conventional cation resins is shortened through decrosslinking which weakens the polymer structure, increases pressure drop, and promotes channeling.

When AMBERLITE 200 was installed in two of American Enka's ion exchange units, its revolutionary polymeric structure was the answer to "aggressive water". Wherever oxidation is a problem, AMBERLITE 200 gives longer resin life than conventional resins, and thus lowers operating cost.

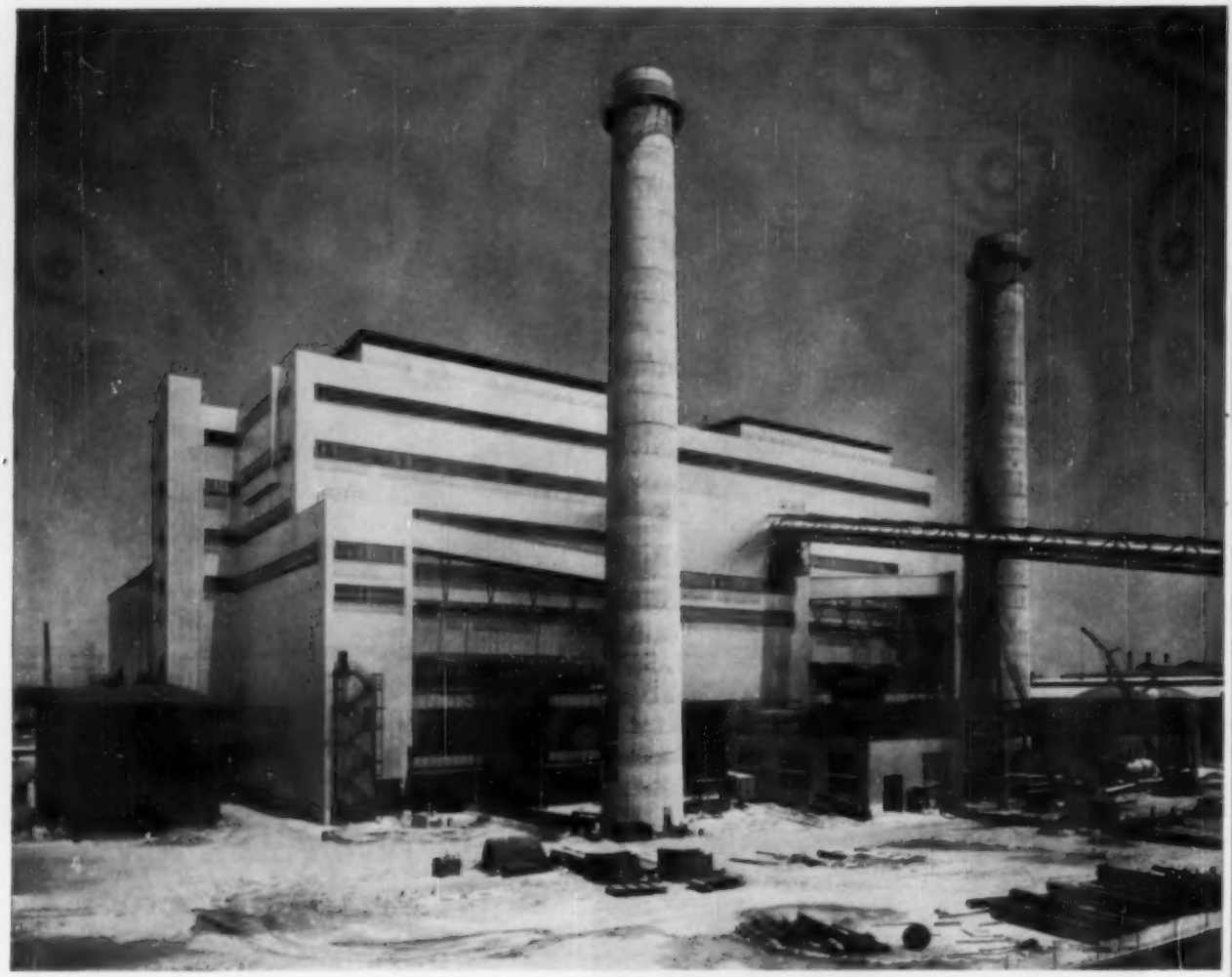
AMBERLITE 200 also has excellent resistance to bead cracking from physical stresses.

For more information on AMBERLITE 200 performance under conditions that degrade conventional resins, write for our technical bulletins. Also ask for a 16-page booklet that shows how many different industries use AMBERLITE ion exchange resins for specific applications—boiler-water softening and deionizing, chemical processing, and many others.

ROHM & HAAS

PHILADELPHIA 5, PA.

AMBERLITE 200



Eddystone Station's Unit 1 has a capacity of 2,000,000 lb/hr at 5000 psig and 1200°/1050°/1050°/F . . . uses C-E Sulzer Monotube Steam Generator.

Vulcan Selective-Sequence system provides Eddystone Station with precision soot blowing

When Unit 1 of Philadelphia Electric's Eddystone Station goes into operation, a Vulcan Selective-Sequence system will accurately control all soot blowing.

Selective-Sequence systems were chosen for both units 1 and 2 at this super-critical station because they assure positive, dependable boiler cleaning . . . make the most efficient use of the blowing medium.

Vulcan Controller saves time. Without leaving the panel, the operator pushes a master button to place the system in sequential operation. He can see that each soot blower is operating in sequence for the proper interval with adequate pressure. He can modify the sequence to improve cleaning or conserve the blowing medium without time-consuming wiring and piping changes.

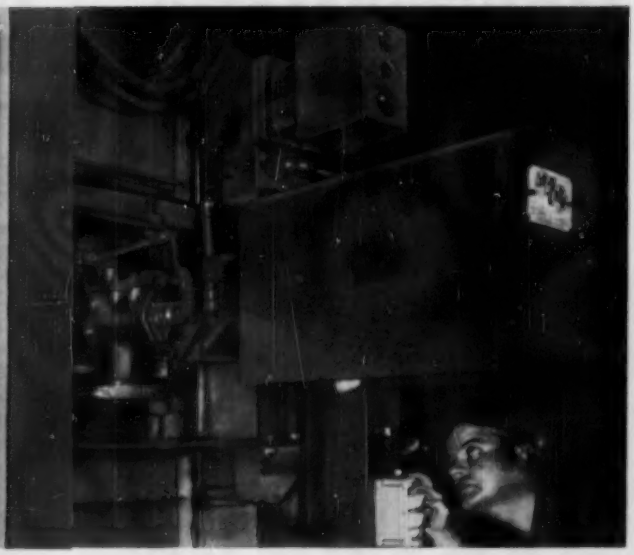
Vulcan long retractables speed cleaning. With dual-motor drive, Eddystone's T-30's minimize the danger of tube cutting or erosion. Low rotating speed increases range and penetration, decreases wear, eliminates whip, and permits cleaning with faster traversing speeds.

Half-'tracts with 19-foot travel, wall deslaggers, and air pre-heater controls are also used. For details, write Copes-Vulcan Division, Erie 4, Pa.

Copes-Vulcan Division

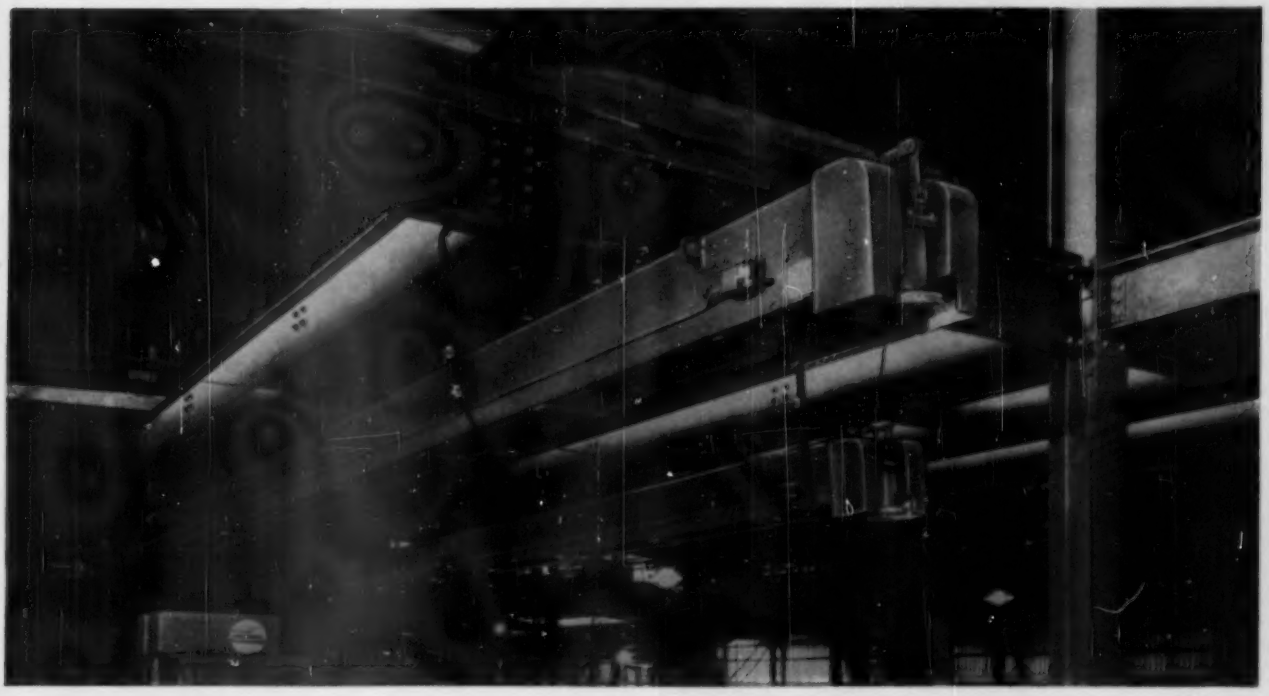
BLAW-KNOX

Panel provides centralized control of soot blowing at Eddystone. With the Vulcan SSC-120 Selective-Sequence controller, each blower can be operated four times during a schedule, or there can be four different schedules. The sequence can be varied by means of a jack board. Any soot blower can be operated at any point in the sequence by plugging one end of a patch cord into the blower jack, and the other end into the desired sequence jack. Write for Bulletin 1029.



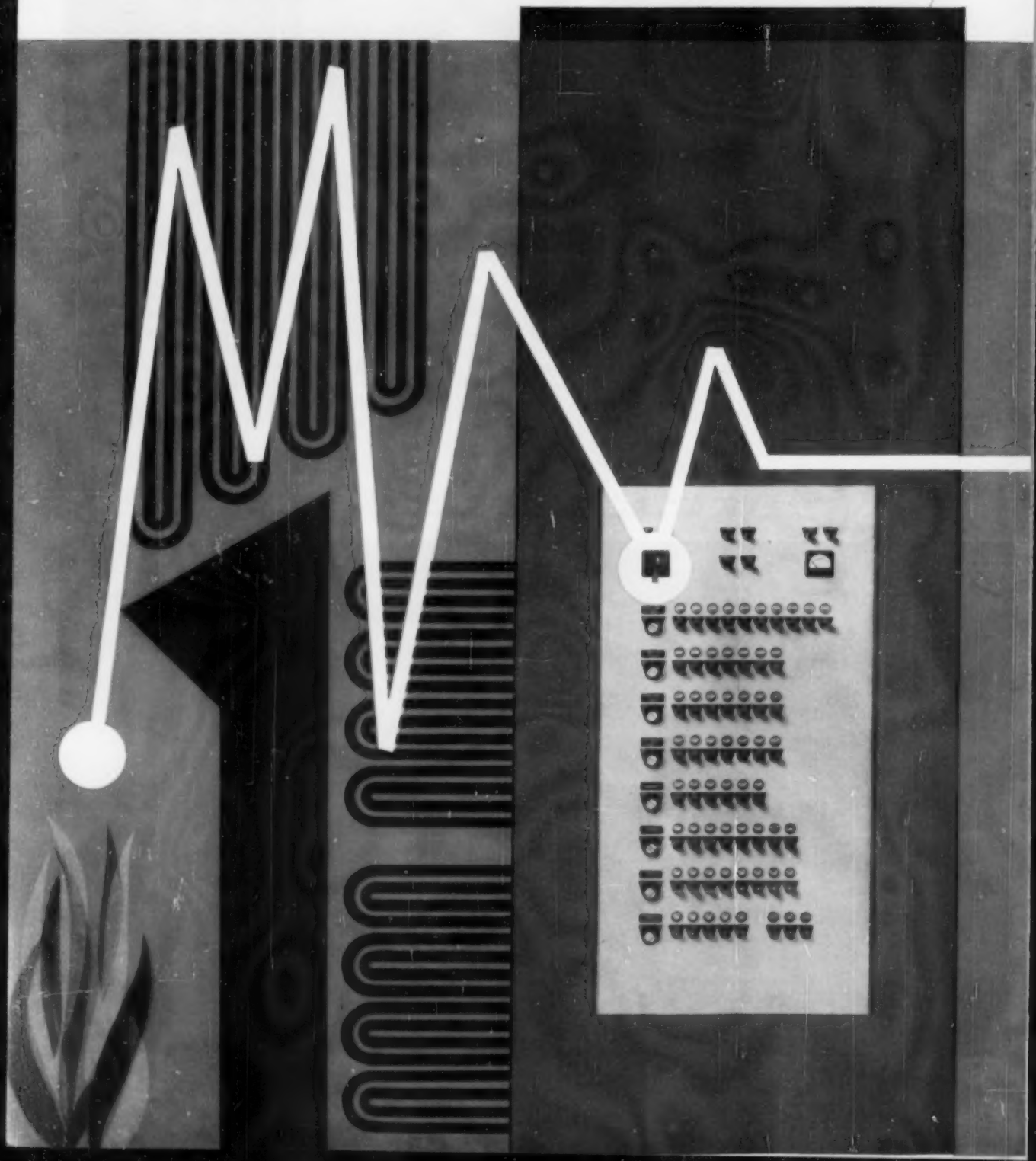
Wall deslagger conserves steam generated at Eddystone by three special package boilers. High striking power of Vulcan RW-3E drives off gummy masses to assure high heat-transfer capacity, and uniform superheat and reheat temperature control. Dual motors are used: one speeds the nozzle to and from the blowing position, the other rotates it slowly for thorough cleaning. All parts are covered for protection, assuring long life with low maintenance. Write for Bulletin 1034.

Vulcan T-30's have 30- and 37-foot travels. Motors are mounted at the boiler end to facilitate maintenance, yet away from heat of the boiler wall. Their placement avoids interferences. Write for Bulletin 1030.



Diamond developed for more economical power

MONITORMATIC...NEW SELF-CONTROL FOR MORE EFFICIENT BOILER CLEANING



Optimum frequency of blower cycling is assured with Monitormatic . . . Diamond's completely automatic boiler cleaning system.

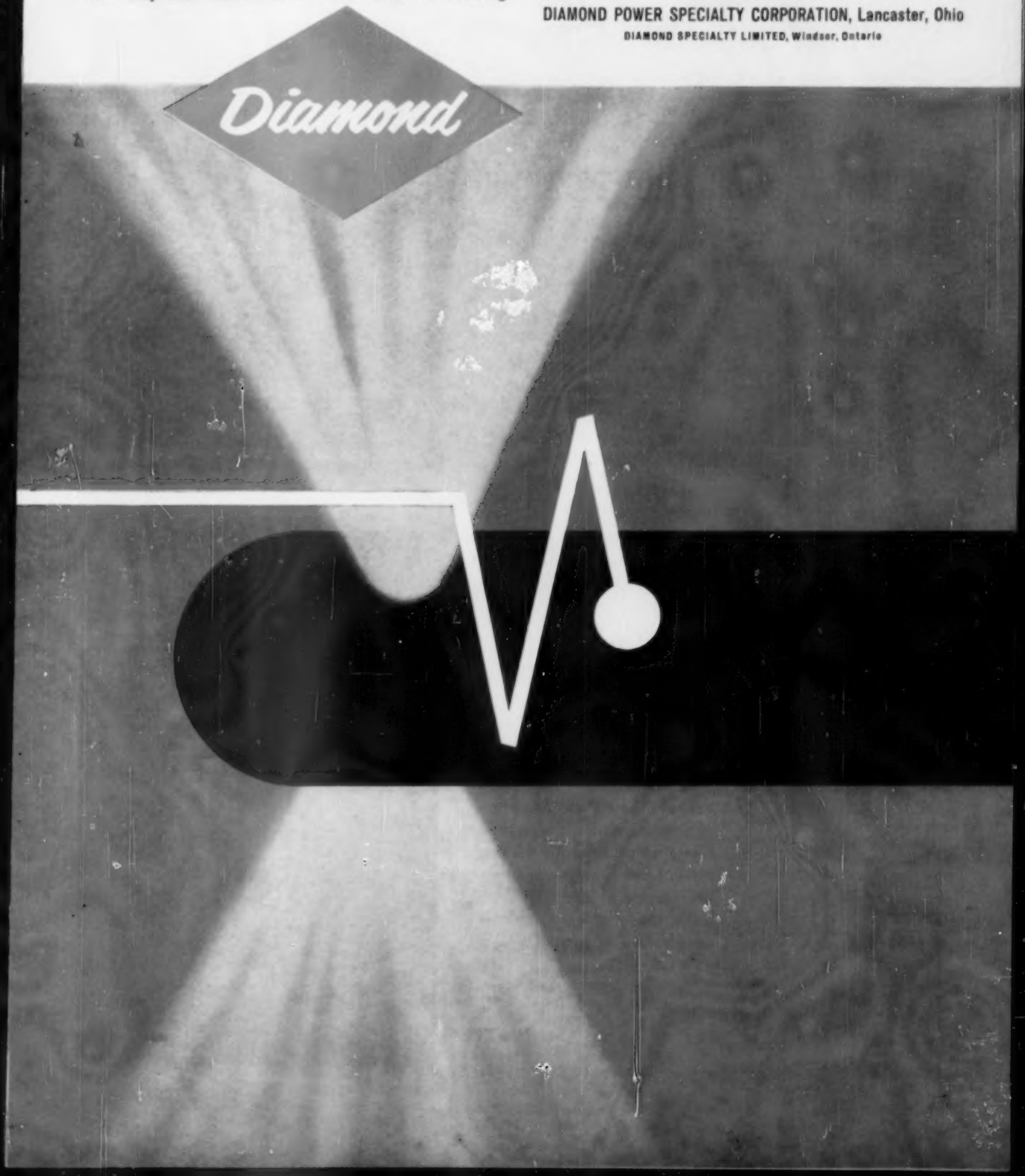
Monitormatic is an automatic method of maintaining constant furnace cleanliness and heat absorption. It responds automatically to changing boiler conditions and eliminates indiscriminate, unnecessary or insufficient blowing. Savings are realized through optimum use of blowing medium, minimized soot blower maintenance, less compressor maintenance if air is used as blowing medium, and reduced operator time. It also automatically adjusts the cleaning system to fuel changes.

Monitormatic is readily and economically applicable to new or many existing Diamond Selectromatic Systems; is another example of why "Diamond Developed" means more economical power generation.

At your request, our Engineers will be pleased to evaluate the application of Monitormatic to your specific system. Call, write or wire for complete information.

DIAMOND POWER SPECIALTY CORPORATION, Lancaster, Ohio

DIAMOND SPECIALTY LIMITED, Windsor, Ontario



YARWAY news briefs

from **Yarnall-Waring Company,** *Philadelphia 18, Pa.*

BRANCH OFFICES IN 19 UNITED STATES CITIES • SALES REPRESENTATIVES THROUGHOUT THE WORLD

HOW YOU CAN SAVE MAINTENANCE...INCREASE OPERATIONAL TIME WITH YARWAY COLOR-PORT GAGES

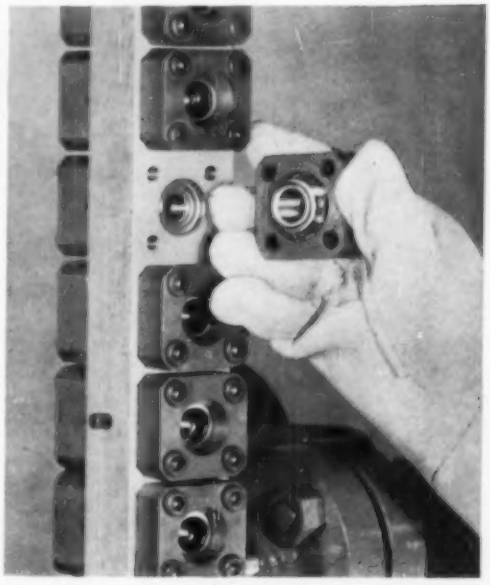
Two big problems confronting the operators of high pressure boilers—excessive maintenance and frequent downtime of water level gages—can be solved with new Yarway Color-Port Gages.

These Color-Port features will help you do the job:

- Patented spring loading of individual port covers, maintains proper pressure on glasses and gaskets at all times.
- Maintenance work can be done *with the gage in place.* NO TORQUE WRENCHES NEEDED!
- Individual "package" port assemblies (glass-mica-gasket) can be replaced in a few minutes.

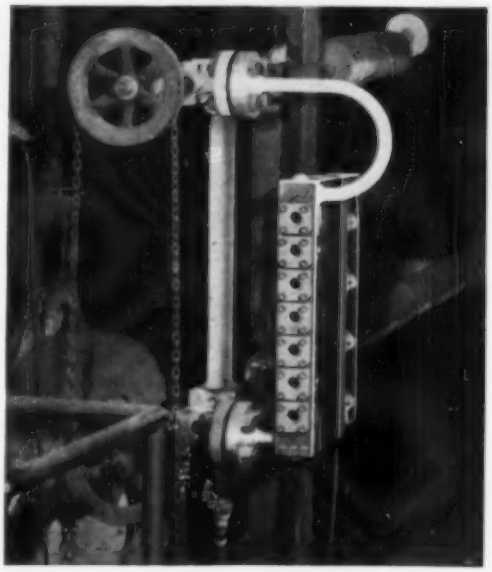
These benefits, PLUS brilliant two-color readings (water space shows green; steam space shows red), PLUS Yarway quality—make COLOR-PORT your best gage buy! 2 Series—1050 psi and 3000 psi.

Write for Yarway Bulletin WG-1815.

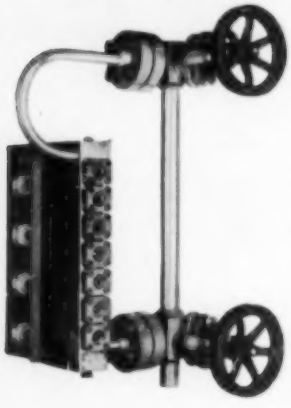


Servicing the Yarway Color-Port Gage is simple! Just remove 4 cap screws (no need for torque wrench), place new "package" assembly in cover, and replace cover assembly. A matter of two or three minutes.

Yarway Color-Port Gage installed on boiler at Alan Wood Steel Co. Leading industrial plants all over the country as well as many major utilities are among the hundreds of satisfied Color-Port Gage users.



NEW COMPACT DESIGN WITH "WELBLOC" VALVES SAVES INSTALLATION SPACE



New "Welbloc" valves on Yarway Color-Port Gages reduce installation space requirements up to 40%. All working parts of valves are easily accessible. Improved direct flow from boiler drum to gage, reduces temperature differential.

This 25' Ljungstrom rotor will contain a total heating surface of 128,000 sq ft. Continuous heat recovery by this unit will boost the temperature of incoming combustion air from 100°F to 585°F. Every 45-50°F thus returned as preheat increases boiler efficiency about 1%.

DOW RECOVERS 425°F WITH LJUNGSTROM® AIR PREHEATER

When Dow Chemical Company's Westside Power House at Midland, Mich., starts up their new 650,000 lb/hr boiler, more than half the waste heat will be *recovered* by a Ljungstrom Air Preheater. Dow's newest Ljungstrom (this is the eighth in their system) was designed to lower stack gas temperature from 700°F to 275°F, thus *recovering 425°F*. As a direct result, company engineers expect at least 10% reduction in fuel costs.

On process stills or boilers—or wherever large volumes of heat are involved—Ljungstrom Air Preheaters improve combustion, make fuel burn more completely. It's possible to save as much as 20% on fuel. ***All* major public utilities** use Ljungstroms; as do many major chemical and petroleum processing companies. Chances are, you, too, could benefit. For additional information, call or write The Air Preheater Corporation, specifying type of application.

THE AIR PREHEATER CORPORATION

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Snow and rain, during the last several months, have combined with fine coal to cause much operating trouble. S-E-Co. Valves, Scales and Feeders are designed to handle the so-called "real estate" type of coal. S-E-Co. equipment will handle the coal all the way from the bunker to the feeder outlet in a stream 24" wide. This large size creates a downward pressure that keeps the coal moving.

The S-E-Co. Twin 24" Coal Valve at the bunker outlet gives two 24" openings. With bad coal, this construction will create twin rat holes in the bunker. This is a weak condition, and the coal has a tendency to cave-in more readily and thus feed.

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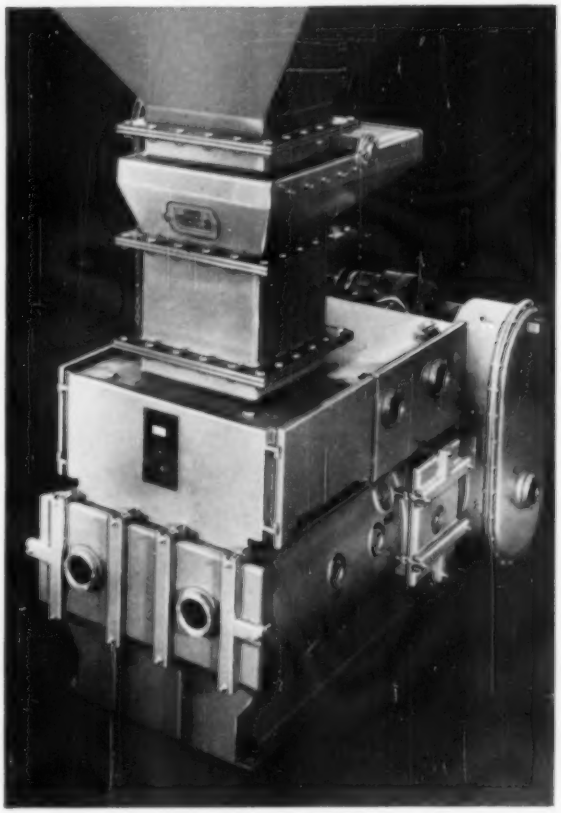
S-E-Co. Twin Coal Valves, Coal Scales, Gravimetric and Volumetric Feeders give you better equipment to cope with the sticking tendency of fine, wet, clayey, low grade coals.

If you have coal flow problems write:



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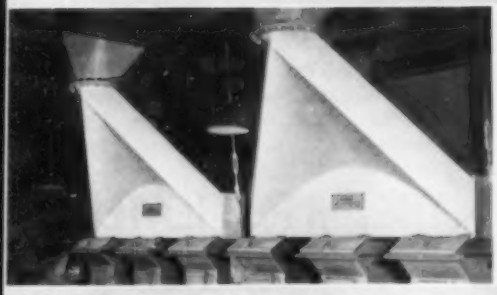

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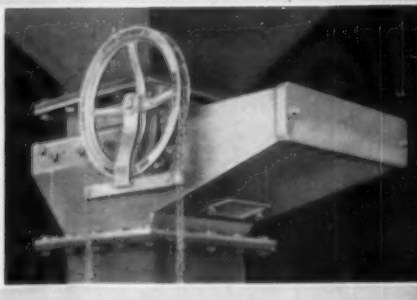

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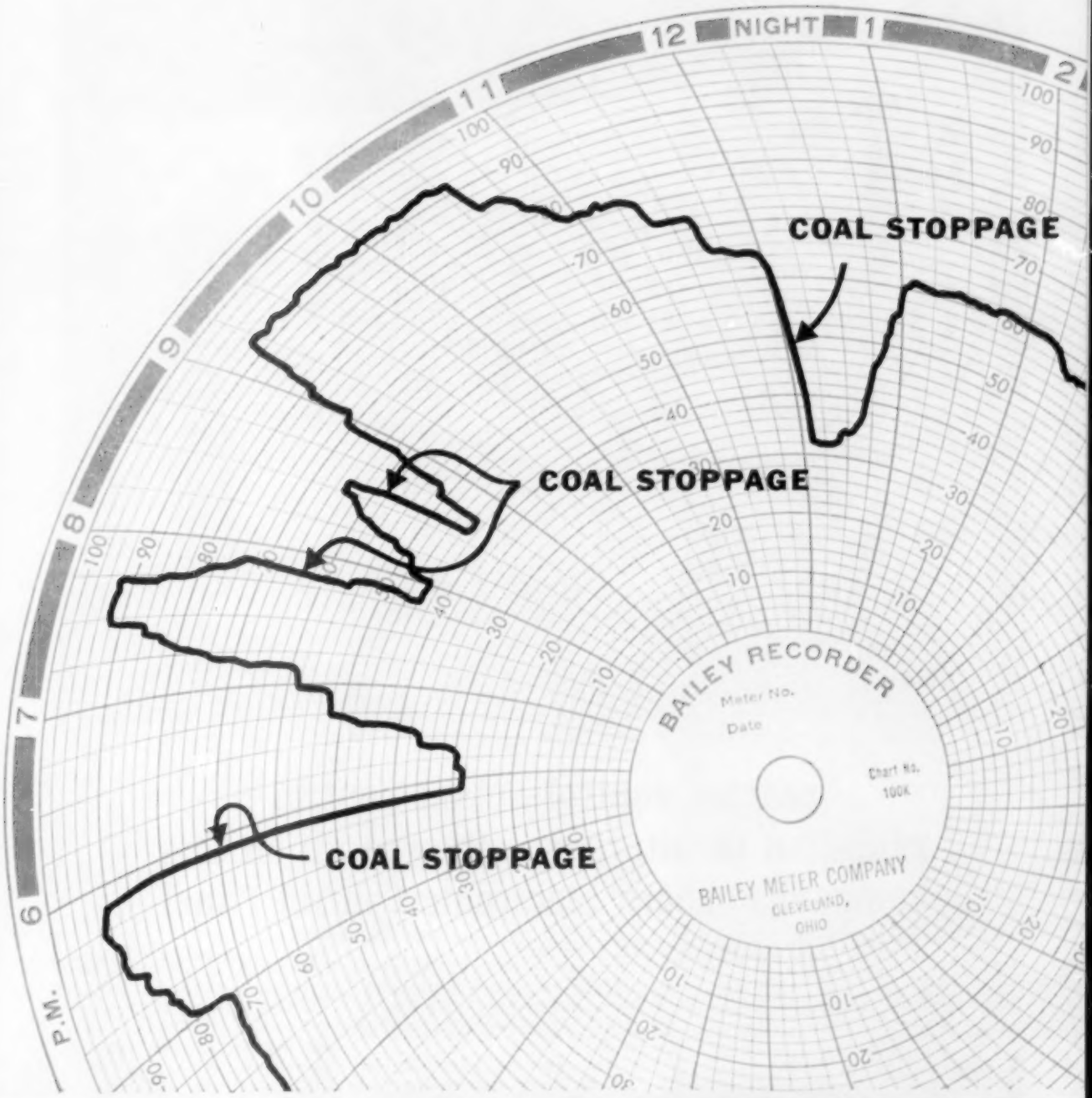

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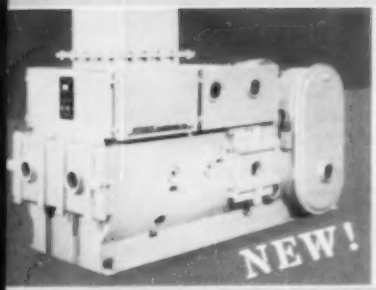

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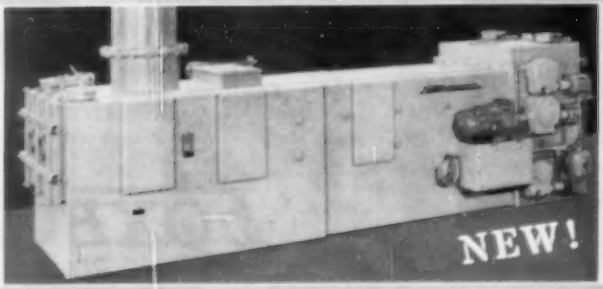

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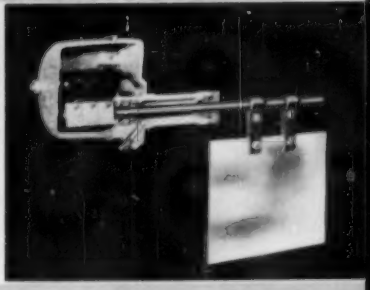
Feed Sticky Coals



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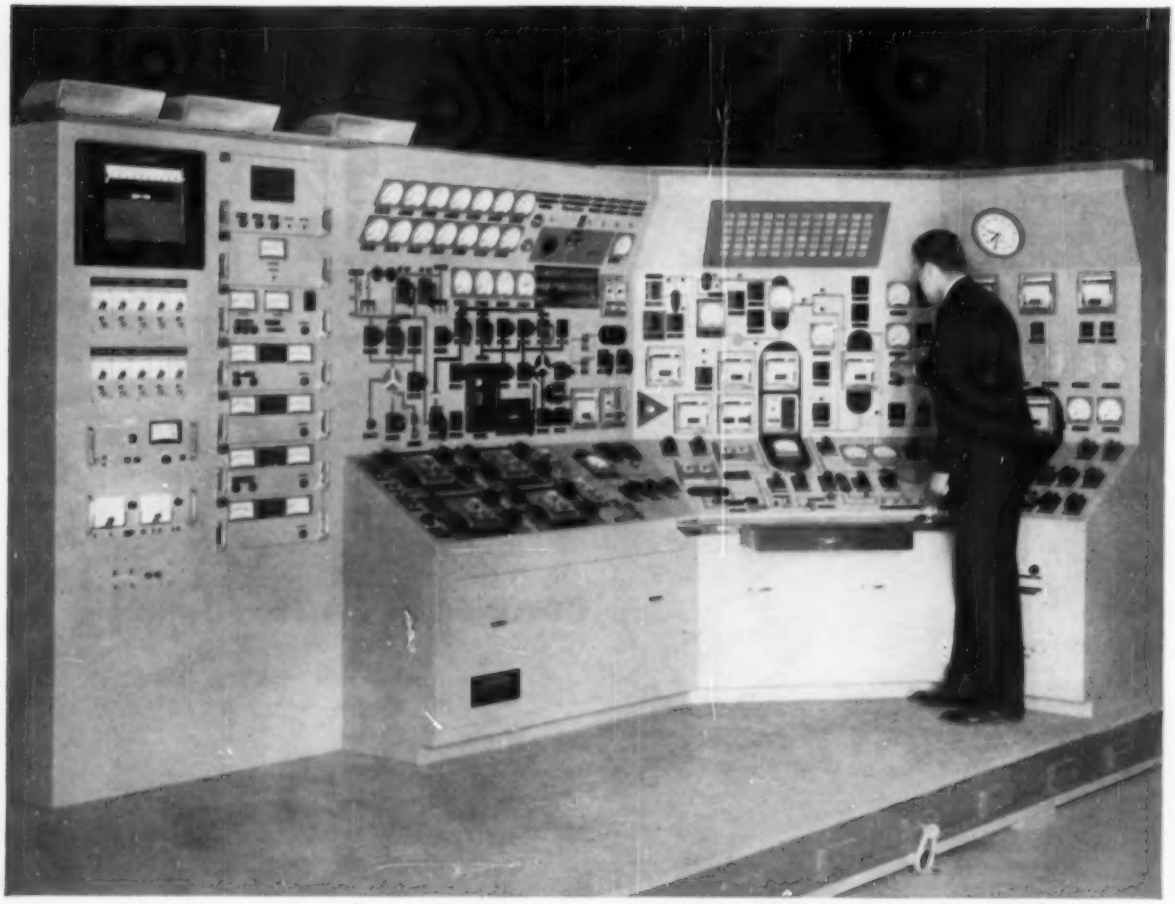

FOXBORO Electronic Consotrol Control Panel for portable nuclear power plant is designed for one-man operation.

reactor control – plant control – radiation monitoring... they're all on this compact Foxboro Electronic Control Panel

SKID-MOUNTED Foxboro Consotrol panel, enclosed in a protective housing, will be "air-lifted" to its Air Force site, 6500 ft. up in the Wyoming mountains.

One skid — that's all the room allowed for control instrumentation on The Martin Company's PM-1 portable nuclear power plant.

The plant — being built for the Air Force under contract to the AEC — will produce both 1000 kilowatts of electrical power and 7 million btu per hour heat for a remote Air Force station high in the Wyoming mountains.

For the PM-1 Reactor, dependable Foxboro Electronic Consotrol* instruments were specified. 100% solid state — no vacuum tubes to replace — instantaneous response. And they're so compact that both nuclear and process control, as well as radiation monitoring, are included on the same operating console.

Compactness — reliability — proven performance — Foxboro Electronic Consotrols are ideal for both nuclear and conventional power plant control systems. Ask your Foxboro Field Engineer to show you the reasons why. Or write for Bulletin 1-17. The Foxboro Company, 275 Norfolk St., Foxboro, Massachusetts.

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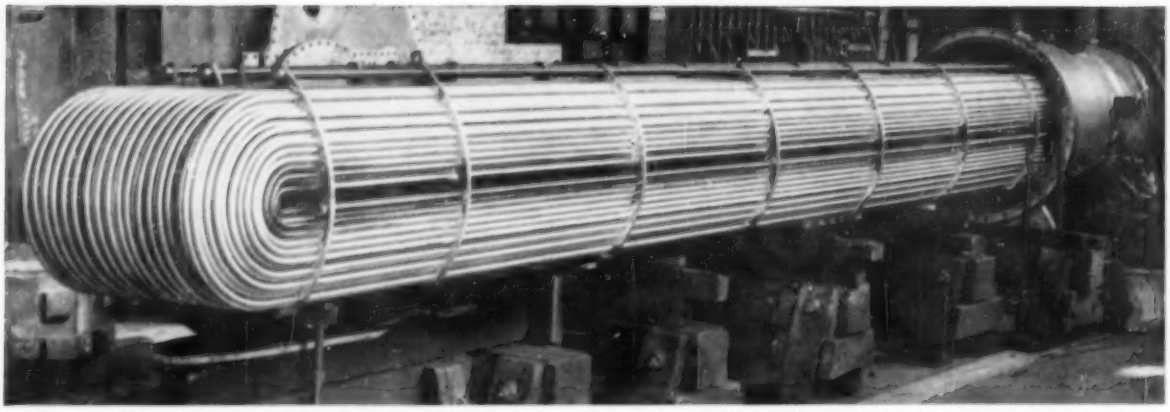

This typical bundle of long-length Anaconda U-Bend tube is the heart of the modern feedwater heater. To get optimum capacity and efficiency—and a more economical and compact unit—designers have lengthened heaters instead of increasing shell diameter.

IN POWER PLANT FEEDWATER HEATERS

Anaconda long-length U-bend tubes offer higher efficiency and greater economy

Feedwater heaters play an important role in the efficient operation of a steam power station—and with today's more demanding conditions, the trend is to the use of long-length U-bend tube bundles in these higher pressure, higher temperature heat exchangers.

U-bend tube design reduces the number of tube joints to one-half. Each U-bend tube takes care of its own expansion stresses. There's no floating head—one less gasketed joint—all resulting in a more economical, more compact heater.

Now Anaconda makes heat exchanger tubes to 100 feet. This means U-bend tubes with legs approaching 50 feet are available. And the radius of the bend can be as small as 1½ times the diameter of the tube—or as great as 30 inches. Dual-gage tubes are made with extra wall thickness in the area of the bends, to cut cost and weight.

With a large family of alloys available for heat exchanger tubes, Anaconda can meet almost every requirement. From Arsenical Admiralty-439 to the cupro nickel group, long-length U-bend tubes can be made to handle a wide range of temperatures and pressures.

Specialists at Anaconda are constantly working with users and manufacturers of heat exchange equipment, helping to solve individual problems. This experience is available to you. For more detailed information on duplex tubes, dual-gage as well as U-bend tubes, see your Anaconda representative. Or write to: Anaconda American Brass Company, Buffalo Division, Buffalo 5, New York. In Canada: Anaconda American Brass Ltd., New Toronto, Ontario. 6061

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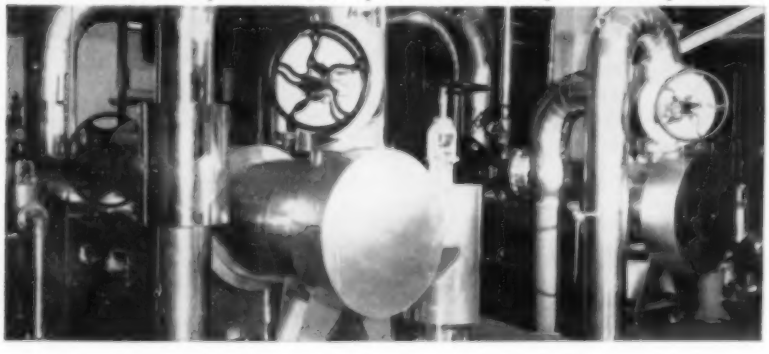
TUBES AND PLATES FOR CONDENSERS AND HEAT EXCHANGERS

Anaconda American Brass Company

Two of the three feedwater heaters at Nichols Station, using a total of 1692 Anaconda U-bend tubes of Arsenical Admiralty-439. Griscom-Russell Co., Massilon, Ohio, designed and built the heaters. U-bend bundles of longer, smaller diameter tubes provide a more effective surface, make possible more compact units. Heaters operate at 200 psi, 450°F.



The new 112,500-kw Nichols Station of Southwestern Public Service Co., Amarillo, Texas.

One way to get heat:

But...Zeigler Superwashed® Coal *does it better!*

Not to mention *easier*—you'd have a Devil of a time efficiently using the flames that Sig. Dante describes in his 14th century "travel book" *Inferno*. But there's really no need—now that famous Zeigler coal is back. It's the same high BTU, low-ash, cleaner-burning coal as the original . . . now even better because it's SUPERWASHED!

Always specify Zeigler SUPERWASHED Stoker Coal for complete satisfaction.

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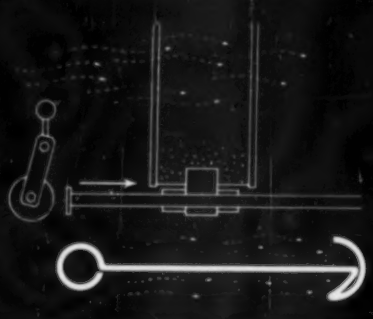
COLLECTING ELECTRODE DESIGN ANOTHER REASON FOR SUPERIOR PERFORMANCE OF BUELL PRECIPITATORS

HEAVIEST CONSTRUCTION, ONE-PIECE FABRICATION Rolled, from a single piece, the structural design and thickness of the Buell collecting electrode provides the strength to retain proper shape and alignment in erection and operation.

EASE OF HANDLING AND INSTALLATION – Rugged sections can be installed without bending or distortion thereby eliminating the possibility of 'arc-overs' (A).

OPTIMUM FLEXIBILITY IN DESIGN OF PRECIPITATOR – Standard 16" width (B) allows increase in size of precipitator by smaller increments (not by 3 foot (A) or longer sections). Permits economical sizing of precipitator to specific requirements of efficiency and space.



POSITIVE "SHEARING ACTION" DESIGN – No reentrainment of dust in rapping. Each row of electrodes is rapped separately–in the direction of the gas flow–on a continuous cycle. Dust is sheared off, drops in an agglomerated mass, and pockets on electrodes minimize reentrainment.

Add to this Buell's exclusive baffling system adjustable after installation for uniform distribution of gas flow, sealed insulators, rigid, 4-point suspension of emitting system, and high emittance, minimum maintenance Spiralectrodes and you have a combination that's hard to beat. You'll be glad you decided on Buell precipitators when you experience superior performance and minimum maintenance. Buell Engineering Company, Inc., Dept. 70-E, 123 William Street, New York 38, New York. Northern Blower Division, 6413 Barberton Avenue, Cleveland, Ohio. Electric Precipitators • Cyclones • Bag Collectors • Combination Systems • Fans • Classifiers

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EDITORIAL

Air Pollution Control—A Joint Responsibility?

A brochure recently published by the John Wood Co. and devoted to the ". . . Blight of Air Pollution" has been called to our attention. This booklet carries the very interesting subtitle of "A Partnership for Local Control" and sets the background for its proposals with a statement accredited to the president of the United States, John F. Kennedy . . . "We need an effective Federal air pollution control program now."

Along with everyone else associated with the power generation field we have watched with misgivings various Federal governmental overtures toward regulation or control of air pollution sources. The John Wood Co. somewhat echoes these misgivings, but it goes beyond a "wait and see" attitude with certain specific proposals. In the main these proposals involve (1) restricting the Public Health Service to its most fundamental function, namely, the proving of the health effects of a contaminated air; (2) establishing a fiscal policy in which both government and industry participate (such things as tax relief, small business loans, research and development funds, since air pollution control and the equipment to effect it represent non-profit expenditures to industry in general); (3) instituting a postgraduate scholarship program for training of competent personnel, the costs again to be shared by government and industry; (4) organizing a central clearing house or sounding board to aid and abet air pollution abatement on a national scale but yet ensure the control and the enforcement authority remaining as a local entity.

Each of the proposals, of course, are subject to considerable pulling and tugging as to their scope, their financing, their application. But in general the plea the booklet makes, "A Partnership for Local Control" is a fair representation of the experiences of enforcement agencies and general industry across the nation where control programs have enjoyed some success.

The traditional U. S. industry approach—namely that each industry clean up its own troubles—does not apply in this instance. Why? We'll let the Wood pamphlet speak for us. "In the air pollution control equipment field itself there are only a few companies large enough to contribute to the research and development of control equipment. By and large, this industry consists of many small and even marginal producers. Consequently, its contribution toward developing more efficient and less costly control equipment, while large relative to the size of the companies, is small when viewed in the perspective of a national program." Amen.

In line with this quotation the booklet's plea for a "partnership" approach deserves a further endorserment when you realize that some of the strongest contributions toward the better application of dust collectors has originated from customer studies such as that reported herein, pages 22–33, by M. J. Archbold of Commonwealth Edison Co. It is our belief that further progress in overall air pollution control from industry sources can come only from the shared experiences of control equipment users and manufacturers.

As the author points out some very basic studies on this general subject were reported by him in 1956 and 1957. In view of the long desired and often attempted set of standards for dust collector performance this most recent report—objective, factual, yet imaginative—is presented here in its entirety because COMBUSTION believes it to be of fundamental value to both the power and process industries as well as the equipment manufacturers

Observations and Experiences Resulting from a Precipitator Improvement Program*

By M. J. Archbold† Commonwealth Edison Company

TABLES I and II based upon Commonwealth Edison System data appeared in the first paper (COMBUSTION Aug. 1956, p 34) and since this information has probably enjoyed a greater popularity than any other part of the paper, it is being extended and reproduced. These tables were made as a guide for the engineer to analyze precipitator problems and manufacturer's proposals. The electrical information was inadequate so Table III has been added. By comparing the data of a precipitator that performs well with one that does not, it may be possible to diagnose the problem. It is also helpful to make the same kind of comparison with a bid in order to judge the proposed performance data.

Precipitators "R" and "T" are added installations tabulated in Tables I, II, and III. These precipitators have been placed in service since the first paper was written and they have given a good account of themselves by producing stack plumes that have been most acceptable and quite clear most of the time.

To amplify the "clear stack" statement, it can be said that it is often possible to approach as close as 300 feet

* Presented at The American Power Conference, Chicago, Ill., March 23, 1961.

† General Mechanical Engineer.

TABLE I—DUST PRECIPITATOR COMPARISON

	Boiler						Gas				Passages					Plate	
	1	2	3	4	6	6	7	8	9	10	11	12	13	14	15	16	17
Collector	Capacity × 1000 (Lb/Hr	Pressure (Psig)	Steam Temperature (F)	R.H. Temperature (F)	Firing	Bottom	Cfm × 1000	Temperature (F)	Velocity (Fps)	Treating Time (Sec)	Number	Width (In.)	Height	Cross Section Area (Sq Ft)	Length (Ft)	Area (Sq Ft)	Area (Sq Ft)/1000 Cfm
A	450	700	758	...	Pulv.	Dry	190		7.0	2.57	34	8	20	452	18	24,500	129
B	450	700	758	...	Pulv.	1-Dry 2-Slag tap	190	360	7.14	1.68	38	8	17.5	443	12	15,960	84
C	500	1300	835	...	Pulv.	Slag tap	270	365	8.77	2.05	44	8	17.5	513	18	27,720	102
D	500	1300	835	...	Pulv.	Slag tap	270	365	8.77	2.05	44	8	17.5	513	18	27,720	102
E	1350	2100	9050	1050	Pulv.	Dry	657.7	305	8.22	1.64	100	8	20	1333	13.5	54,000	82.2
F	500	1325	935	...	Pulv.	Slag tap	200	350	5.13	2.34	54	8	17.5	630	12	22,800	114
G	600	1325	955	...	Pulv.	Slag tap	275	350	6.37	2.84	54	8	20	720	18	38,880	141
H	730	1900	1050	...	Cyclone	Slag tap	384	385	9.1	1.32	60	8	17.5	700	12	25,200	65.5
I	1100	1900	1050	1000	Cyclone	Slag tap	546	334	8.15	2.2	84	8	20	1120	18	60,500	110
J	425	1325	935	...	Pulv.	Slag tap	211	350	7.76	2.32	40	8	17	453	18	25,200	119
K	375	1275	910	...	Pulv.	Slag tap	180	360	8.57	1.4	30	8	17.5	350	12	12,700	70.5
L	750	1325	935	...	Pulv.	Slag tap	285	340	6.78	2.65	60	8	17.5	700	18	37,900	133
M	600	1325	935	...	Pulv.	Slag tap	270	350	7.15	1.68	54	8	17.5	630	12	23,100	84.4
N	400	1325	910	...	Pulv.	Slag tap	210	350	7.32	2.46	41	8	17.5	479	18	25,800	123
O	1320	1900	1050	1010	Cyclone	Slag tap	624	340	12.0	1.33	72	8	18	864	16	41,472	66.5
P	1800	2100	1050	1050	Pulv.	Dry	780	287	6.95	2.59	68-In 68-Out	8.25, 8	20	1813	18	92,160	118
Q	1108 1030	650	755	...	6-Pulv. 1-Cyc.	6-Slag tap 1-Dry	1450	340–350	6.32	2.53	300	8.5	18		16	172,800	119
R	1450	2000	1050	1050	Pulv.	Dry	690	300	6.03	3.32	108	9	23.5	1903.5	20	101,520	147
S	2100	2000	1050	1000	Pulv.	Dry	1116	300	6.26	2.87	108, 156	8.25, 8	24	2970	18	148,600	134
T	2200	2000	1050	1000	Pulv.	Dry	1058	290	6.57	3.04	152	9	23.5	2680	20	142,880	135
U	2055	2000	1050	1000	Pulv.	Dry	953	290	6.08	2.96	148	9	23.5	2610	18	125,200	131.5
V	2076	2000	1000	1000	Pulv.	Dry	1051	284	5.84	3.60	150	10	24	3000	21	151,100	143.8
W	2046	2000	1050	1000	Cyclone	Wet	964	280	6.97	3.44	128	9	24	2300	24	147,200	153
X	750	1325	935	...	Cyclone	Slag tap	305	385	5.9	3.05	48	9	24	864	18	41,472	136
Y	3712	2400	1000	1000	Pulv.	Dry	1553	286	6.92	2.89	208	9	24	3744	20	199,700	128.5

before the plume becomes visible or before a visitor can be certain that the unit is operating.

The "Committee on Improvement of Existing Precipitators" studied the data on precipitators "R" and "T" to determine the alterations that might be made on other units which were not performing so well. Precipitator "L," reported in Tables I, II, III, was one unit which we had reason to believe was not operating efficiently, and yet you will note that its gas velocity, plate surface, and wire length compares very favorably with "R" and "T."

There are two very significant items which were not tabulated in Tables I and II. The first is gas and dust distribution and the second is power to the electrodes.

Fig. 1 shows the general layout of precipitator "L." The duct turns at the inlet and outlet are very sharp and the installation has been crowded into a very limited space, thus preventing the installation of the usual straight run of ducts before and after the precipitator. It was the general consensus that good gas and dust distribution would be extremely difficult to obtain. Subsequent field tests, with the boiler cold, showed that gas distribution was moderately good. The tabular air heater preceding the precipitator, undoubtedly, contributed to this beneficial condition.

Fig. 2 shows the velocity in a representative gas passage in the inlet of this precipitator, which is almost as good as the distribution obtained on precipitators "R" and "T" under similar test conditions.

Since the gas and dust distribution was acceptable, the next step was to investigate the amount of power. There were two mechanical rectifiers operating half-wave serving four bus sections on each of these two boilers. It was a simple matter to reconnect the rectifier for full-wave operation on one outlet bus and to connect the spare full-wave on the other. Point "B" on Fig. 3 shows the efficiency obtained with this arrangement, which is quite an improvement over the original arrangement as shown by point "A."

Since there was no sparking, it was logical to think that even more power could be used. It was for this reason that four (fully automatic, tube type, 460 ma, 50 kv) rectifiers were borrowed from a unit under construction. The results were astonishing, considering that this precipitator has only two fields in series and that these tests were run without benefit of wire rapping. It was necessary, however, to increase the plate rapping to maximum intensity during light load periods in an attempt to maintain efficiency. After a week's run, the build-up on the wires was fully $1^1/_2$ in. in diameter and the turning vanes were also quite dirty. These two items accounted for some of the reduction of efficiency.

The results of these tests are shown as points 1 to 10 inclusive, on Fig. 3 where efficiency versus watts per 1000 cfm of gas is plotted. Fig. 4 shows the grain loading versus watts per 1000 cfm of gas for the same tests.

Table IV shows the maximum loading of current and watts per linear foot of wire, per square foot of plate, and per 1000 cfm of gas obtained during the tests on precipitator "L."

During the test, the inlet and the outlet were both operated at 460 ma which was full rating of the sets. The output of the two sets on the outlet sections was increased briefly to 615 ma, causing only moderate sparking. This experience was sufficient to prove that larger sets could be used. The next size set rated at 615 ma has, therefore, been shown in the tabulation in Table IV.

Until more tests of a similar nature are run on other precipitators, the test figures in Table IV will be used as a bench mark, or guide. This is now being done in studying the data in Tables I, II, and III. Since there are many variables involved some judgment must be used in making the comparisons, but such comparisons will give the engineer a good idea of the expected performance for precipitators listed in Tables I, II, and III, which are not performing properly.

As a result of these tests, two full-wave, 460 ma sets for the inlet sections, two full-wave, 615 ma sets for the outlet sections, four additional plate rappers, eight wire rappers, and turning vane soot blowers have been in-

TABLE II—DUST PRECIPITATOR COMPARISON

Bus Section				Wire					Type			Plate Rappers				Wire Rappers					
18	19	20		21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
Collector	Number	Sect /100,000 Cfm	Plate (Sq Ft)/ Section	Quantity (Ft)	Spacing (In.)	Ft./Bus Section	Ft./1000 Cfm	Efficiency Guarantee (%)	Collector	Rectifier	Wave	Type	Number	Length of Cycle (Sec)	Rapped at One Time (%)	Plate (Sq Ft)	Type	Number	Length of Cycle (Sec)	Rapped at One Time (%)	Wire (Ft)
A	3	2.1	6,125	16,350	9	4,087	86	95	Elec.	E	Half	M-I	16	...	6.25	1530	...	4	...	...	4090
B	4	2.1	3,990	...	...	...	...	90	Elec.	M	Half	Motor	4	...	25	3990	...	0	...	...	...
C	6	2.22	4,620	24,640	5.25	4,107	91	92.5	Elec.	M	Half	M-I	8	30	12.5	3460	...	0	...	...	...
D	6	2.22	4,620	24,640	5.25	4,107	91	92.5	Elec.	M	Half	M-I	8	30	12.5	3460	...	0	...	...	...
E	12	1.83	4,500	34,080	7	2,840	51.8	97	Comb.	E	Half	M-I	12	60	...	4500	Syn.	12	...	...	4028
F	4	2.0	5,700	18,900	6	4,725	94.5	92	Elec.	M	Half	Motor	4	60	25	5700	...	0	...	...	...
G	4	1.45	9,720	34,560	5	8,640	125	95	Elec.	E	Half	M-I	16	60	6.25	2430	...	0	...	...	...
H	4	1.04	6,300	21,000	6	5,250	54.6	80	Elec.	M	Half	Syn.	18	360	5.55	1400	Vib.	24	...	...	875
I	8	1.46	7,560	53,760	8	6,720	98.5	90	Elec.	E	Half	M-I	16	30	8.3	3780	Pneu.	8	...	...	6720
J	4	1.9	6,300	16,300	7	4,075	77.3	95	Elec.	M	Half	M-I	4	10	25	6300	Syn.	4	...	...	4070
K	4	2.22	3,175	11,550	7	2,887	64.2	90	Elec.	M	Half	Pneu.	4	90	25	3180	...	6	...	...	...
L	4	1.4	9,475	33,230	6	8,300	120	90	Elec.	M	Half	M-I	8	12	25	740	Syn.	8	...	...	4170
M	4	1.48	5,700	18,900	7	4,725	70	90	Elec.	M	Half	Pneu.	8	...	12.5	4890	...	0	...	...	...
N	6	2.86	4,300	21,500	6	3,583	102	95	Elec.	M	Half	M-I	24	60	4.17	1070	M-I	6	...	...	3580
O	6	1.06	6,912	33,200	10	5,533	58.5	90	Elec.	M	Full	Pneu.	12	360	8.33	3370	...	0	...	...	...
P	8	1.026	11,520	85,000	...	10,625	109	97	Elec.	E	Half	M-I	32	...	12.5	2880	...	16	...	...	5310
Q	24	1.66	7,200	124,800	9	5,200	86	95	Elec.	M	Full	E-M	240	...	...	719	Pneu.	72	...	...	1730
R	20	2.9	5,076	60,912	6,12	3,045	88	98	Elec.	E	Half	M-I	64	...	1.56	1585	Vib.	32	...	...	1990
S	18	1.62	8,260	136,000	6	7,550	122	98	Elec.	E	Half	M-I	36	...	2.78	4130	M-I	36	...	2.78	3780
T	20	1.89	7,144 avg.	85,760	10	4,290 avg.	81.1	98	Elec.	E	Half	M-I	64	60	3.12	2230	Syn.	32	...	6.24	2680
U	20	2.1	6,260	83,500	9	4,175	87.7	98	Elec.	E	Half	M-I	48	60	2.08	2610	Syn.	24	600	...	3480
V	24	2.28	6,300	100,800	7.5	4,200	96	98	Elec.	S	Half	M-I	72	60	1.39	2100	Syn.	48	300	4.2	2100
W	16	1.66	9,200	98,300	10	6,140	102	98	Elec.	S	Half	M-I, G-I	48	120	2.08	3070	Syn.	32	600	3.12	3070
X	8	2.62	5,184	27,648	7, 10, 5	3,456	90.6	97.5	Elec.	S	Half	Syn.	20	240–900	5.0	3080	Syn.	16	240–900	6.25	1730
Y	32	2.06	6,240	119,800	7,	3,750 11.63	77.2	98	Elec.	S	Half	M-I, G-I	96	120	2.08	2080	Syn.	64	300–600–900	3.12	1087

E—Electronic tube. E-M—Electro magnetic. M—Mechanical. S—Silicon.

TABLE III—DUST PRECIPITATOR COMPARISON

Collector	Kva per Set	Rated Output (Ma)	Rated Output (Kv), RMS	Rated Output (Kw/Set)	Ma/Plate (Sq Ft)	Ma/Wire (Ft)	Ma/1000 Cfm Gas	Watts/Plate (Sq Ft)	Watts/Wire (Ft)	Watts/1000 Cfm Gas	49 Bus Sections in Series	Number of Sets
A	15	230	50	10.35	0.0188	0.0281	2.42	0.845	1.27	109	2	2
B	25	250	50	12.5	0.0335	0.0415	2.63	1.68	2.07	132	2	2
C	25-I	250-I	75	12.5-I	0.018-I	0.0203-I	2.04	0.903-I	1.01-I	104	2	I-I
	15-O	150-O		7.5-O	0.0216-O	0.0243-O		1.085-O	1.21-I			2-O
D	25-I	250-I	75	12.5-I	0.018-I	0.0203-I	2.04	0.903-I	1.01-I	104	2	I-I
	15-O	150-O		7.5-O	0.0216-O	0.0243-O		1.085-O				2-O
E	15	230	50	10.4	0.0256	0.0405	2.11	1.15	1.82	94.5	3	6
F	25	250	75	12.5	0.0219	0.0264	2.50	1.09	1.32	125	2	2
G	25	250	75	12.5	0.0129	0.0145	1.82	0.643	0.724	91.0	2	2
H	25	250	75	12.5	0.0198	0.0238	1.30	0.992	1.19	65	2	2
I	25	250	75	12.5	0.0165	0.0185	1.83	0.827	0.93	91.5	2	4
J	25	250	75	12.5	0.0204	0.0307	2.37	1.02	1.53	118	2	2
K	15	150	75	7.5	0.0236	0.0261	1.67	1.18	1.30	83.2	1	2
L	25	250	75	12.5	0.132	0.0150	1.75	0.66	0.75	87.8	2	2
M	25	250	75	12.5	0.0216	0.0265	1.85	1.08	1.32	93.5	2	2
N	25	250	75	12.5	0.0291	0.0349	3.57	1.46	1.74	178	3	3
O	25	250	75	12.5	0.0362	0.0452	2.65	1.81	2.26	132	1	6
P	70	800	45	36	0.0348	0.0377	4.10	1.56	1.70	185	2	4
Q	35-I	350-I	75	17.5-I	0.0415-I	0.0338-I	3.52	2.07-I	1.68-I	177	2	6-I
	25-O	250-O		12.5-O	0.0592-O	0.0481-O		2.96-O	2.40-O			12-O
R	20	290	50	14.5	0.0229-I	0.0381-I	4.2	1.03-I	1.715-I	189	4	10
					0.0458-O	0.0762-O		2.06-O	3.43-O			
S	40	474	50	21.34	0.0286	0.0313	3.85	1.291	1.410	172	3	9
T	30	460	50	20.7	0.0258-I	0.0428-I	4.35	1.16-I	1.93-I	195	4	10
					0.0515-O	0.087-O		2.32-O	3.86-O			
U	30	460	50	23.0	0.0367	0.055	4.83	1.65	2.48	218	3	10
V	50	700	53	37.5	0.0557	0.083	8.00	2.49	3.73	357	4	12
W	45	615	50	27.7	0.0335	0.0502	5.10	1.50	2.26	230	4	8
X	35-I	500-I	50-I	25-I	0.0483-I	0.0723-I	7.88	2.17-I	3.25-I	354	4	4
	50-O	700-O	50-O	35-O	0.0676-O	0.101-O		3.04-O	4.56-O			
Y	45	615	50	27.7	0.0492	0.082	6.31	2.22	3.68	284	4	16

* In calculating the watts on wires, plates and in flue gas, the voltage drop through the tube or mechanical rectifier has been deducted.
I—Inlet Bus Section O—Outlet Bus Section

stalled on one precipitator with very satisfactory results. The rectifiers are operating at full capacity most of the time. The moderate sparking indicates that the rectifier capacity installed was a good choice.

At the present time it is not known whether the power input to the precipitator was limited by the amount of gas, plate, wire, or dust load, or whether it was a combination of all four.

Up to this point the paper has been concerned principally with electronic rectifiers, whereas, the Commonwealth Edison system operates a larger number of precipitators using mechanical rectifiers. A considerable

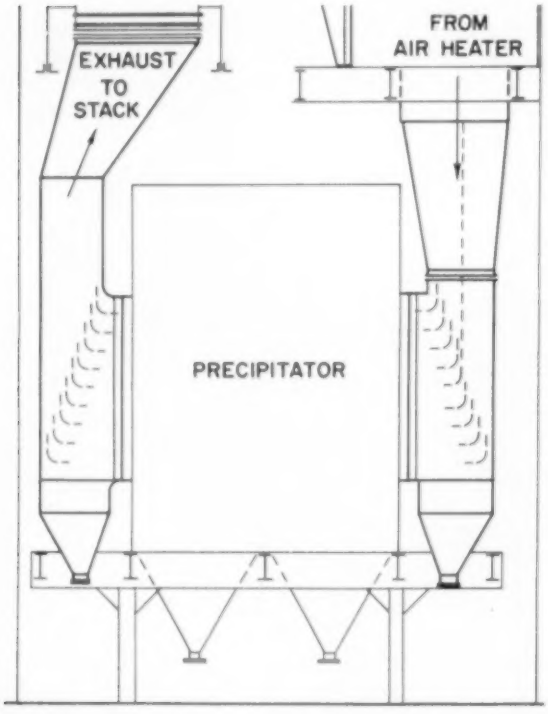
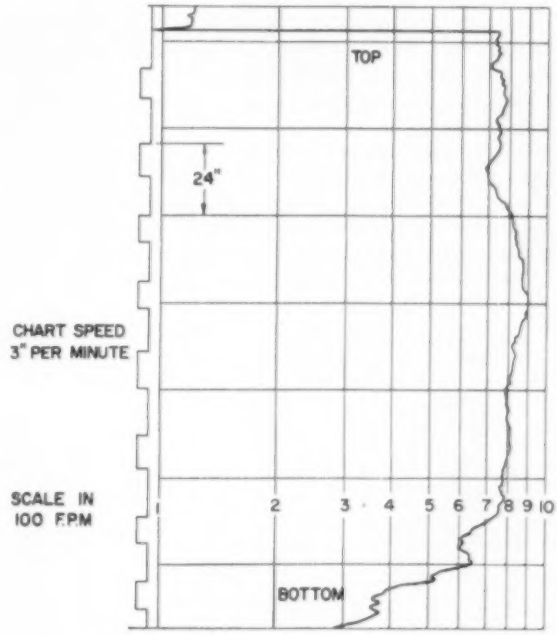
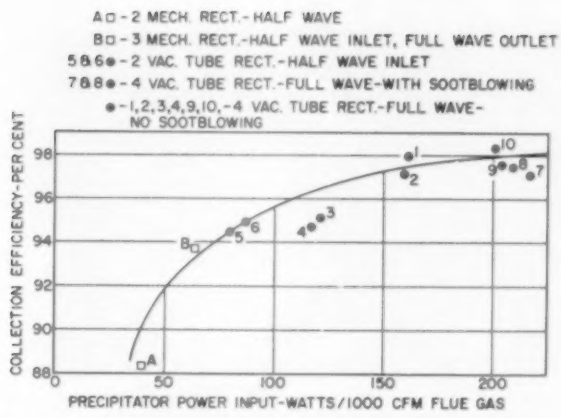


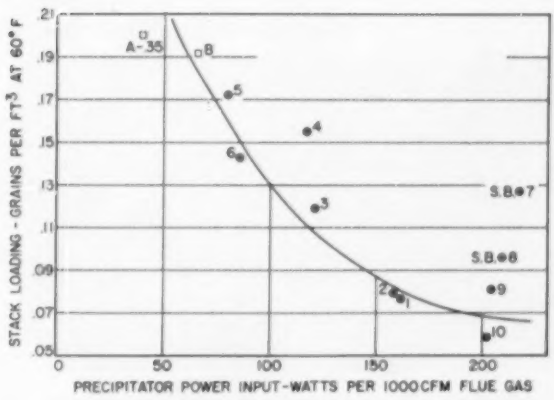
Fig. 1.—General layout of precipitator "L," Tables I, II, III, shows the right angle turns of the duct work and the cramped quarters in which it was installed



The Curve Represents the Air Velocity in Passage #1-A of the Precipitator, and it Was Obtained by a Curve Drawing Instrument, Actuated by a Hot Wire Anemometer, Guided by the First 2 Wires in the Passage. The Left Hand Block Curve Shows the Position of the Sensing Element as it was Raised by a Motor Driven Drum From the Bottom to the Top of the Collector Plates.

Fig. 2.—Representative gas passage in the inlet of precipitator "L" produced the above air velocity pattern which indicates a reasonably good distribution


A□ - 2 MECH. RECT. - HALF WAVE
B□ - 3 MECH. RECT. - HALF WAVE INLET, FULL WAVE OUTLET
5&6● - 2 VAC. TUBE RECT. - HALF WAVE INLET
7&8● - 4 VAC. TUBE RECT. - FULL WAVE - WITH SOOTBLOWING
●-1,2,3,4,9,10,-4 VAC. TUBE RECT. - FULL WAVE - NO SOOTBLOWING
COLLECTION EFFICIENCY - PER CENT
PRECIPITATOR POWER INPUT - WATTS/1000 CFM FLUE GAS



Figs. 3-4—Performance curve above for precipitator "L" plots efficiency against power input. Point A represents a hookup with the two original mechanical rectifiers operating half-wave; Point B is after reconnection to put one rectifier full wave on one outlet and its spare, full wave on the other. Points 1 to 10 are plot points for a series of tests run employing four fully automatic rectifiers. Fig. 4, right, above, shows stack loading in grains versus power input for these same tests

amount of work has been done to obtain better performance from the mechanicals, such as operating with a closer gap setting and at a slightly higher output. Since they do not readily lend themselves to automatic voltage control and are usually operated well under their optimum ratings to prevent frequent trip-outs, further efforts toward improving collection efficiency through their use has been deferred and the efforts are now being directed toward the use of automatic voltage controlled electronic rectifiers for increasing collection efficiency.

Table V shows test data obtained on precipitators "C" and "D" mechanical rectifiers. This is the first time Edison has obtained precipitator input and tabulated the results.

In comparing these results with those in Table III, it will be noted that they are only one-fourth to one-third as great. It might be possible to make up some of this difference by installing more or larger mechanicals, but it is questionable whether this would be advisable since they are not automatic and would probably operate considerably below the rating required.

In turning attention to precipitators "H" and "I", Tables I, II, and III, which are next on the test schedule to use high capacity electronic sets, a somewhat different problem is found. These precipitators are on cyclone-fired furnaces which produce possibly half as much dust as pulverized-coal furnaces, but the sizing of this ash is materially finer so that the power requirement can conceivably be much greater. In addition to this, the plate surface on "H" is only half of that on "L." Likewise, the length of plate in direction of gas travel is 30 per cent less and the gas velocity is considerably higher.

To determine whether more power could be added, preliminary tests have been run by making temporary connections from a boiler out of service for overhaul, so that four mechanical rectifiers, one for each field at full-wave, was operating instead of the two rectifiers serving the four fields at half-wave. This meant the power input to the rectifiers was doubled and this additional power reduced the precipitator outlet dust loading in half. It will be very interesting to learn how much more power these relatively small precipitators will be able to accept.

Gas Distribution Experiences

Three dimensional models for gas distribution tests have been included in all contracts for precipitators since the purchase of "Q." Precipitator inlet duct design for "R" and "T", Tables I, II, III, followed very closely the designs indicated by scale model tests. These ducts and turning vanes have performed very well when clean.

Fig. 5 shows the dust build-up on vanes in the entrance elbow to precipitator "I" and is an excellent example of conditions prevailing where no provisions have been made for cleaning. The polish marks on the side wall shows the jet action produced by the restricted inlet vane area. This elbow is now being equipped with soot blowers. The next two figures show the outlet elbow of precipitator "E." Fig. 6 is a view of the access door, showing the extent of the dust build-up in the elbow. The dust covers the bottom turning vane and is approximately six feet deep at the distribution plate. Since not less than one-third of this plate was covered it can be assumed that this blockage seriously affected the gas distribution.

Fig. 7 is a view of the interior through this same door. This figure shows the dust build-up on the upper vanes and also the large amount on the outlet distribution plate. Plans are under way for the installation of soot blowers to keep the elbow and vanes clean. Model tests are being made to determine the value of the outlet distribution plate. If it is not needed, it will be removed;

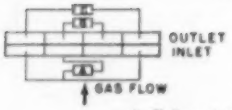
TABLE IV—TEST DATA—PRECIPITATOR "L"

	Current (Ma)			Power (Watts)		
	Per Ft of Wire	Per Sq Ft of Plate	Per 1000 Cfm Gas	Per Ft of Wire	Per Sq Ft of Plate	Per 1000 Cfm Gas
Outlet	0.0742	0.065	6.72	2.78	2.45	272
Inlet	0.0554	0.0486	6.72	2.45	2.16	272

				Actual Voltage	Kw
Outlet	Ao | Bo	A—615 ma	B—615 ma	V = 37,400	46.2
Inlet	Co | Do	C—460 ma	D—460 ma	V = 44,500	40.94
					87.14

Gas Flow ↑ Gas—320,400 cfm Plate surface, 37,900 sq ft
Effective wire length, 33,320 ft

TABLE V—MECHANICAL RECTIFIERS, PRECIPITATORS "C" AND "D"



	A-C Input to HV Transf.				D-C Fed to Pptr.				Current (Ma)			Power (Watts—d-c)			
	Amp	Volts	Kva	Kw	Kv	Center (Kw)	Outside (Kw)	Total (Kw)	Wire per Ft	Plate per Sq Ft	Gas per 1000 Cfm	Wire per Ft	Plate per Sq Ft	Gas per 1000 Cfm	Current (Ma)/ Rectifier
Inlet	62.6	286	17.8	13.7	30	3.66	3.66	7.32	0.0198	0.0176	214*	0.595	0.528	64*	244
Outlet	34.7	316	10.95	9.28	29.9	2.52	2.45	4.97	0.027	0.024		0.805	0.717		166

* These figures are based on contract gas flow. The actual gas flow is higher, but it was not measured during this test.
Performance on test, December 1959. Gap spacing, $^1/_{16}$ in.

Rating
A—25 kva — 250 ma
B & C—15 kva — 150 ma } 50,000 v
270,000 cfm gas*
8.77 fpm gas velocity*

27,720 sq ft plate surface
24,640 ft wire

but if it is, rappers will be installed to keep it clean.

It has been thought that only air should be used for cleaning dust from areas preceding and following the precipitators. It is, therefore, interesting to know that 150 psig steam has been used as a blowing medium for the past two years on "R." An inspection recently made of the inlet elbow after a nine-months run showed the vanes and also the floor of the elbow ahead of the precipitator clean. Some dust had accumulated beyond the range of the blower. The dust in the elbow was dry and fluffy, showing no evidence of moisture. It has been concluded from this experience that steam soot blowing can be used without deterimental effect on the dust. It should be pointed out, however, that the steam must be dry when entering the soot blowing element.

Pressure on the element is far more important than the blowing medium. Precipitator "T" is a good example. Ninety psig service air has been used for cleaning the ducts on "T" and this pressure is much too low. As a result, four to five feet of dust has been deposited in the center of the inlet duct. Such dust build-ups must upset the distribution pattern considerably. Consideration is now being given to obtaining a higher pressure source for this operation, and it could be either steam or air, depending upon piping cost and other charges.

A very good example where air has been successful is on "P" shown in Fig. 8. Without soot blowers the turning vanes in the elbow immediately below the regenerative air heaters were very heavily loaded with dust. The bottom five were completely covered and four feet of dust was piled up in the center of the duct. The balance of the vanes were more than half full. The 150 psig air-puff type soot blowers, which were subsequently installed, are doing an excellent job of keeping both vanes and ducts clean.

The gas distribution in ducts following the regenerative type heaters has been quite poor. Since turning vanes turn the gases given them quite faithfully, the poor distribution patterns from the heaters persist beyond the turning vanes. In "T" this condition has been greatly improved by placing two perforated distribution plates in the duct ahead of the precipitator.

The need for a perforated plate on the outlet of the precipitator still appears to be open to question by some in the precipitator industry. The outlet plates have been omitted from several Edison precipitators without any apparent ill effects. In these installations considerable effort has been made to obtain uniform horizontal gas flow entering the precipitator with the expectation that the gas will continue to flow in a like manner to the outlet, providing that there are no internal obstructions to upset the pattern. Proof of this reasoning is found in "T" which has two perforated plates on the inlet and none on the outlet. The scale model studies of this precipitator showed good results and these have been corroborated by field tests.

Further evidence of the effectiveness of the second perforated inlet plate was just recently developed in a two dimensional model study on "P." The poor gas distribution on this precipitator was described and shown in model tests in the 1957 paper, COMBUSTION, April, 1958, p. 34. At that time, good distribution was obtained in the model by covering the two hoppers and by installing a second set of vanes at the bend line where the duct begins to enlarge to meet the precipitator opening.

Recent field tests showed the velocity in the duct leaving the air heater to be four times as high near the back side of the duct as it was on the other side. Model studies just recently completed duplicated the gas velocity as shown in the field tests, and it was indeed gratifying to find that a perforated plate placed $3^1/_2$ feet in front of the entrance plate (this places two in series) gave a very good pattern in the precipitator model. The positioning of this plate in the model was quite critical. Since it will not be necessary to install the second set of turning vanes, a considerable amount of money will be saved.

It is interesting to reflect on the change in thinking during the past five years. At the time the 1956 paper was being prepared it was hoped that the vanes could be so designed that perforated distribution plates could be entirely eliminated. It took some time before it became apparent that vanes could not correct poor distribution patterns, but merely direct them, and that reliance must still be placed on perforated plates.

The specifications now being written request two inlet distribution plates, but none on the outlet.

Referring again to the model test of "P" precipitator, it was acknowledged that the two perforated inlet plates were making a good gas distribution pattern out of a poor one, but it was still questionable whether the dust would distribute as well as the gas. By placing strips of scotch tape a short distance behind the precipitator entrance it was possible to collect samples of dust introduced ahead of the air heater outlet elbow. The quantity of dust collected on the tapes was quite uniform, indicating that the dust does follow the gas quite faithfully at these velocities.

The design of the perforated plate or plates, relative to per cent of free opening and hole size, is more im-

Fig. 5—A five-year accumulation of dust build-up on turning vanes of the entrance elbow to precipitator "I"

Fig. 6—Access door in outlet elbow of precipitator "E" shows dust build-up there

portant than we had realized. Model tests on perforated plates with various ratios of free area indicated an improvement as the percentage of free area is increased with an optimum improvement at about 50 per cent. Model and field tests on "E," which is the combination mechanical-electrostatic collector with a 35 per cent free opening distribution plate between the mechanical and electrostatic collector, indicated that this amount of free area actually over-corrected the gas distribution. This condition was corrected to give a 51 per cent free opening by cutting out the ligaments between every other pair of holes. Model tests showed the performance of this plate to be equal to a new 51 per cent plate. At this time we are considering only 50 per cent free area distribution plates with $1^1/_4$ in. to $1^1/_2$ in. diameter holes. The smaller holes should produce less objectionable high velocity gas jets in the precipitator lanes and hence, create less wire vibration and fewer trip-outs.

Beyond this, our experience indicates that the inlet opening to the precipitator should be completely filled with these plates, that the perforations should be uniformly distributed, that there should be no other obstructions to upset a good distribution pattern, that the plate should be engineered for ease of rapping and be equipped with rappers, and that the flow discharging from these plates be uniformly distributed.

To more fully explain the above paragraph, perforation diameters range from $1^1/_4$ in. to 4 in. Plates are often not perforated to the edge of the sheet, but left with enough undrilled material to seriously obstruct the flow through the passages located directly behind the junction of two plates. Many installations have a gap between the plate sections and as much as eight in. at the bottom which permits the blasting of gas into the collector. Still others have support beams that will block off areas as much as twelve in. in width at several locations across the precipitator.

Experiences on Turning Vanes

In the 1956 paper, mention was made of the need for keeping turning vanes clean and of several methods being used. The first method used syntron vibrators which were attached to a specially prepared section of turning vanes. After more than six months' operation inspection showed that the syntrons removed little or none of the dust accumulation. The experiment was discontinued.

Another section of vanes were equipped so that they could be rotated periodically to dislodge the ash. The vanes were mounted on a shaft which was supported with bearings on each end. A number of shafts were connected so as to allow several of the vanes to be rotated at the same time. Almost immediately after starting the boiler the vanes "froze" and could not be rotated. After several attempts had failed this experiment was abandoned.

Another section was built with small vanes placed fairly close together in order that the gas velocity would sweep them clean. This test was not successful, largely due to locating the vanes in a low velocity section of duct. Other vane installations have since been made which have been successfully cleaned by the gas velocity, but they have been in sections of the duct where the velocities are about 40 fps and where there was no expansion of gas in passing through the vanes. Velocities of this magnitude do not normally exist in the region near the precipitator, therefore, high gas velocity is not normally a good solution.

Soot blowers installed to clean turning vanes have been successful and it is this method which is now being used for cleaning vanes as well as deposits in the ducts.

Corrosion and Insulation

Corrosion of the interior of the shell has become more of a problem as steam-generators become more efficient and hence, produce a lower exit gas temperature. A number of the pulverized-coal units on the Edison system are operating with flue-gas temperatures as low as 270 F. This is at the SO_2 dew point or slightly below.

To eliminate the corrosion or to reduce it to a minimum, it is necessary to keep the interior surfaces above the dew point and this calls for a good insulating job. It was thought that an all welded penthouse above the precipitator would obviate the need of insulating the top of the precipitator shell, since the heat radiation into the penthouse should keep the shell roof temperature above the dew point. In two recent installations (precipitators "R" and "T") the shell roof insulation was omitted. After one year of operation the shell roofs had corroded badly resulting in iron rust peeling off in large sheets. "T" has recently been insulated with $1^1/_2$ in. magnesia block which in turn is covered with $^1/_2$ in. Marinite board to provide a good walking surface. One and one-half in. magnesia block with a trowelled

plaster finish was also applied to the flowerpot insulators. Although this insulation was dried out prior to being energized, it still flashed over on all flowerpots and had to be removed. Therefore, thermal insulation on flowerpots is not recommended. Recently the shell roof of precipitator "R" was insulated with $1^1/_2$ in. calcium silicate blocks covered with $^1/_4$ in. mineral wool cement, $^1/_8$ in. mastic cement, and $^1/_{16}$ in. asphalt asbestos felt. The I-beams on the roof were covered with $^1/_8$ in. cork-filled mastic. The access doors were covered with a 2 in. glass wool blanket which is laid over the doors and overlapping one ft onto the roof insulation.

The records show that precipitator "G," which went into service in September 1953, has had exit gas temperatures as low as 225 F at night and on week ends while burning gas at light loads, and as low as 250 F while burning coal under similar conditions. At full load the temperature is often 265 F. These conditions make this installation quite interesting from the corrosion standpoint.

The precipitator is located on the boiler-house roof where it is subjected to the cold winter winds of northern Illinois. The shell and all the breechings are covered with magnesia block insulation and weatherproofed with Insulcote. The precipitator roof is insulated with $1^1/_2$ in. magnesia block, covered with one-half in. Marinite board. A recent inspection showed no corrosion either on the duct, precipitator walls, or the roof. The doors which had a trowelled insulation on the inside had corroded through, requiring patching several times and finally replacement. The top corners of the precipitator, where gallery attachments conducted the heat from the interior metal, were heavily corroded. From this experience it can be concluded that to avoid corrosion the metal must be kept above the acid dew point.

Fig. 7—Interior view of outlet elbow, Fig. 6, indicates heavy dust build-up on turning vanes and perforated distribution plate, left

Another approach to this problem is in precipitator "P" which does not have a tightly welded penthouse, but has a well ventilated "Butler" building that serves the same purpose. The roof of the precipitator has $1^1/_2$ in. magnesia block with a trowelled coat over the top. Steel plates are used for walkways to the principal equipment. There is no noticeable corrosion on the interior of this precipitator after three years of service at low temperatures. In many ways this type of protection is superior to the all welded penthouse. The "Butler" building serves as a housing for the rappers, rectifiers, and bus ducts; it also permits maintenance to be done on this equipment during all kinds of weather and at any hour of the day. About the only objection to it is its higher cost.

Flowerpot insulators require a certain amount of air circulating through them to prevent internal dust build-up. This same movement of air into the flowerpots prevents the diffusion of SO_3 out of the precipitator and into the penthouse. In a tightly welded penthouse it then becomes necessary to provide several ventilating openings to supply the air necessary for the flowerpots.

On precipitator "G" the penthouse vents were closed with pipe plugs. It did not take long for the SO_3 to diffuse into this area with consequent condensation and a very heavy corrosion attack on the penthouse walls. This drippage also was the cause of flowerpot insulator flash-overs and failures.

These examples should provide assistance to those looking for answers to their corrosion problems.

Relationship of SO_3 and the Precipitator

In relation to the low level economizer study the SO_3 content of the gas was measured ahead of and following precipitator "E." The results of these tests are tabulated in Table VI. It will be noted that in the tabulation the SO_3 content of the gas is reduced slightly as the gas passes through the air heater and the reduction is con-

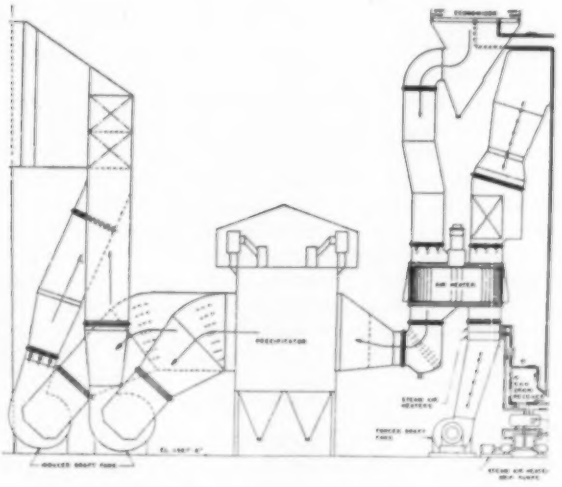


Fig. 8—Turning vanes (90 deg.) leaving air heater direct gas to precipitator "P." Gas goes from there through 45 deg. turning vanes to a run of duct entering at a 90 deg. turn to the horizontal. Gas leaves ID fan discharge and then through a final 90 deg. turn to the stack. Without soot blowers the dust build-up becomes impassable at the air heater discharge turning elbow vanes and the center of the duct-work following

TABLE VI—SULFUR TRIOXIDE CONTENT OF FLUE GAS

Precipitator "E"

Temperature (Deg. F)	Avg. SO_3 Ppm by Volume — Air Heater — Inlet	Air Heater — Outlet	Ppt. — Outlet
650–700	22.9	...	...
325–335	...	21.9	12
290–300	...	9.4	4.3
280–285	...	7.1	2.0

These results are averages of many readings, from a number of test runs, taken over a period of 6 days.

siderably more as it passes through the dust collector even at the higher temperature. It is logical to assume that the reduction at the higher temperature is due to some constituent of the dust, such as magnesium or calcium, combining with the SO_3. The completeness of such a combination is a function of the quantity of the constituent, the time of contact, and the mixing or turbulence of the gas and dust.

At the lower temperatures (280–300 F) it is believed that the SO_3 gas is chilled to form a mist and either adheres to, or reacts chemically with the dust, since the SO_3 content has been reduced considerably in going through the air heater. The additional reduction that takes place in passing through the precipitator could be attributed to the SO_3 being precipitated as particulate matter. Whatever the action, the reduction of SO_3 as a gas is quite marked at the lower temperatures.

If these tests raised some questions, they also answered several questions. If the precipitator was located between the air heater and the low level economizer, and if the temperature of the gas was maintained just below the point where SO_3 gas becomes particulate, the SO_3 would be for all practical purposes removed. It would then become necessary to keep the temperature of ducts, economizer parts, and stack steel above the SO_2 dew point, or 120 F to keep from corroding these metals.

You might say then that it would be beneficial to permit the gas to drop to as low a temperature as possible in the air heater. So far as corrosion is concerned that is probably true, but there are other factors that need to be considered and they are as follows:

A few years ago, temperature regulation was considered to be of prime importance in avoiding plugging of the regenerative type air heater. Edison's designs called for a 225 F cold end temperature of the air heater. To obtain this temperature meant t' at the air to the heater would be 150 F and the gas leaving would be 300 F.

For the past few years several units have been operating as low as 200 F cold end temperature. Air enters the air heater at 130 F and the gas leaves at 270 F. With adequate soot blowing equipment operated a high percentage of the time it has been possible to keep the air heaters in service. Also it has been possible to operate the mechanical collectors without forced outages. Even though both the air heater and mechanical collectors have been washed twice in 5 years, and this has been expensive, the electrostatic collector can be in the most difficulty with this sticky dust.

To handle the increased power needed for greater precipitator efficiency, collector plates, and more especially the high tension wires must be kept cleaner than had previously been considered necessary. This means that the rapping systems for both the plates and the wires must be engineered and constructed much better to remove the ash that clings so much more tenaciously. A large part of the trouble experienced on recent installations has been due to inadequate rapping systems. To make plate and wire cleaning a bit more effective and easier, it is hoped that the gas temperature can be maintained between 280–290 F at all loads on the two jobs equipped with low level economizers and to let the economizer reduce the stack temperature to 225 F and lower.

Gas Sneakage

The following paragraphs point out the locations where gas sneakage may occur, and they elaborate on the magnitude of the problem and suggested remedies.

Gas sneakage in dead passages in several precipitators on the system was found to be roughly 200 fpm during cold air tests at 50 per cent velocity. Under full load conditions this velocity would possibly be 400 fpm or more. In one particular precipitator the cross-sectional area of the dead passages amount to eight per cent of the total. It can readily be seen from this that unless these dead passages are well sealed at both the inlet and outlet an appreciable amount of gas will pass through untreated.

The sneakage above the collector plates on one precipitator was found to be 300 fpm during a 50 per cent velocity cold test. Since there is a considerable area above these plates, this represents a fairly large quantity of untreated gas.

This precipitator has three fields in series. In order to reduce gas sneakage at this point, it is planned to install a baffle plate between the roof and the collector plate support beams. One baffle could be placed between the first and second field and possibly another between the second and third field. It is planned to determine the actual number needed by model test.

Gas sneakage in the hoppers of one 98 per cent collector reduced the efficiency during tests to 95 per cent. By installing lights and observation ports under the collector plates, considerable dust re-entrainment could be seen coming from under the hopper baffle. Re-entrainment continued until the hopper filled sufficiently to form a seal. The extension of the baffle plates reduced the gas sneakage and re-entrainment, which resulted in the precipitator performing as guaranteed. Hopper sneakage not only permits gas to pass through untreated, but it also re-entrains dust already collected, thereby considerably increasing the dust loading.

One other source of trouble which is not classified as gas sneakage, but which does upset precipitator performance is the excess air leakage which may come into the penthouse and leak through the top of the flowerpot insulator bushings into the precipitator. These covers should be made so that this leakage can be regulated to allow just enough leakage to prevent build-up of the dust inside the bushings and no more.

Access to the Precipitator Interior

Until recently, the need for access to the precipitator interior on the Edison system was not fully appreciated. Some large precipitators do not have planned access and these are extremely difficult to inspect, clean, or maintain. With the use of higher corona power and lower gas temperature, the tendency to collect dirt on

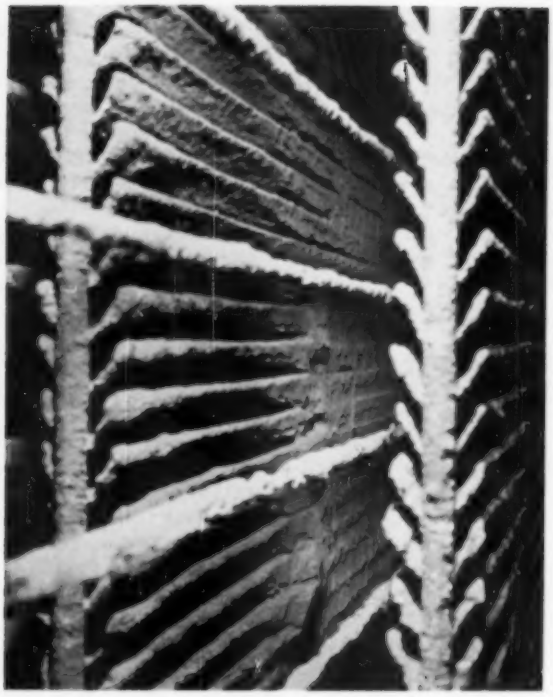


Fig. 9—Precipitator "P" V-pocket electrode, showing the trowelled-on appearance of the dust build-up in the pockets

Fig. 10—Precipitator "E." View from above the plates, looking down toward the entrance, showing "do-nuts" on the high tension wires

both wires and plates is much greater and consequently, it has been necessary to hand clean during outage. These experiences have indicated that it is highly desirable to have good access to all parts of the precipitator.
(*Editor's Note: Although the author does not make the reference the illustrations, Figs. 5-10, underline this point.*)

None of the precipitators on the system have any means provided for gaining access to the interior from the entrance. Serious consideration is being given to putting hinged sections in the perforated plate immediately ahead of the precipitator.

Access through the interior from the sides is also highly advisable. Whether one or two passages are required will depend upon the number and the arrangment of the fields in series, and of the length of treatment. In one precipitator, ladders placed at the shell wall make it possible to make very comprehensive inspections. Walkway grating near the bottom of the collector plates will also be needed in most installations.

There have been several occasions where it has become necessary to straighten plates or wire frames, and in those cases where access through the interior was not available, it was impossible to do this maintenance without removal of the high tension wires. This certainly is not a desirable situation.

There are several collectors on the system having no head room above the plates, whereas, others have a bare minimum. Those precipitators which do have ample head room offer excellent working space for cleaning, inspection, and maintenance of this area.

All that has been said previously also holds for the area below the plates and at the outlet. For those precipitators which have no distribution plate at the outlet, it is usually very easy to gain access to the interior.

Automatic Voltage Control

Fig. 3 shows the relationship between precipitator efficiency and the power input per 1000 cfm of gas (with the gas flow held constant throughout the test). It is apparent that the efficiency is an increasing and smooth function of the watts per 1000 cfm of gas.

The ampere contribution to the watts is not directly controllable, but is itself dependent upon the applied volts which can be controlled. When these are mathematically combined the efficiency of a precipitator has classically been shown to be a function of the voltage squared. However, many authorities feel that it is a function of at least the voltage cubed. The importance of sustained high voltage is apparent. Control would be quite simple if there were no sparking between the discharge wires and the plates. Such is not the case; however, it is the duty of a good automatic control to maintain the highest possible voltage (and thus, efficiency) consistent with the spark rate and hold the short circuit current associated with the sparks to reasonable levels.

It has been shown by tests that a precipitator is most efficient if the voltage is raised until there is something approaching 100 sparks per minute.

It should be mentioned that the voltage for any given sparking level varies widely with gas temperature, dust, and moisture conditions, particularly at start-up and shut-down. During a day's operations, the voltage might vary 10,000 to 15,000 volts.

The only control that has to date appeared completely satisfactory is one in which a series wound saturable reactor that is rated for full line voltage is put in series with the precipitator step-up transformer. The characteristics of the series wound saturable reactor are such that the load current through the saturable reactor is a

function of the control signal to it and is independent of conditions in the precipitator. The short circuit associated with sparking current is only slightly more than load current.

It is now common for the saturable reactor to receive its control signal from any one of a number of types of spark sensing devices. While in operating detail the various spark sensing circuits work differently, all appear to provide a satisfactory signal for the saturable reactor.

It is now our desire to have the automatic control lower the precipitator voltage for all troubles in the precipitator and have the main circuit breaker trip out only for failures in the power supply. Several collectors on the system are equipped with automatic voltage control, and they are rendering very satisfactory service.

Miscellaneous

V-Pocket Electrodes

There are six precipitators using this type of electrode on the system. The first 2 were installed a number of years ago when the flue-gas temperature was normally 375 F. At this high temperature the dust remains dry and when rapped from the plates it falls between the pockets in a quiescent zone and is presumably not re-entrained in the gas stream. They apparently gave satisfactory service at this temperature.

Recent installations, however, operated approximately 100 F below that temperature and the dust is not as dry, but tends to become sticky. The V-pocket type plate does not perform as it was intended. Fig. 9 shows the dust build-up between the pockets where it has hardened sufficiently to present a problem in dislodging the dust. A build-up of this type cannot possibly fall in a quiescent zone, but will bounce out in the gas stream and aggravate the re-entrainment problem. The V-pocket electrodes is not suitable for dusts of this character.

"Do-Nuts" on High Tension Wires

It may be difficult for those who have not had the opportunity of observing the interior of a precipitator to visualize the dust build-up on the wires commonly referred to as "do-nuts." The "do-nuts" on the wires in Fig. 10 are quite moderate in size and have formed in spite of the continual use of syntron vibrators. On precipitators where no vibrators are installed, these deposits often exceed four in. in diameter. This deposit affects the power input to set significantly and varies in relation to the resistivity of the ash and its thickness. In the pilot plant experiments recently conducted, the power input to the electrodes dropped 30 per cent in two hours of operation without plate or wire rapping. The total thickness of wire and dust at that time did not excess $^{3}/_{8}$ in. (wire was 0.109 in. diameter).

Efficiency

To make certain that dust collectors will meet the guaranteed efficiency during regular operation, the specifications call for the precipitator to be operated for two weeks without any adjustments or alterations prior to the acceptance tests. In addition to this, other normal operating functions should be continued during the tests, such as, operation of rappers and soot blowers and dust hopper unloading.

Inductive Interference on Voltage Control Circuits

In the temporary rectifier installation for precipitator "L" considerable difficulty was encountered during the start of the tests due to induced interference between the high voltage cables. The power and voltage is sufficiently high to transmit the sparking indications from one cable to the other and to consequently, upset the controls. These cables or feeds should be shielded or placed far enough apart so that the inductive effect will be reduced to a minimum.

Effect of Structural Members on Gas Distribution

In making field velocity tests, large irregularities in the velocity curves were noted. These were usually caused by structural beams running across the precipitator. One case was traced to the magnetic impulse rapper bars. These four rappers were mounted on the side of the precipitator and were located one above the other, so that the four flat bars effected quite a wide path in the gas distribution pattern. If possible, neither the ducts nor the precipitator should have any cross members in the gas stream.

Instruments

Instruments supplied on mechanical rectifier installations were usually an a-c ammeter and voltmeter. With such meager instrumentation very little information relative to the actual daily performance or to the possible maximum power input to the precipitators was obtained. Although, it was possible to make rough estimates concerning the power applied to the high tension electrodes, this was rarely done.

With the installation of the first automatic voltage controlled electronic rectifiers, there also came the d-c milliammeters, voltmeters, and spark rate meter. It not only became possible for the operator to determine for the first time such important guide posts as milliamperes per 1000 cfm of gas, per square foot of plate surface, and per lineal foot of wire, but also a still more valuable bench mark; watts per unit of gas, plate and wire. The spark rate meters usually read zero and this was probably the first concrete evidence that the precipitators were underpowered. Based upon actual field experience, precipitator collection efficiency continued to improve as power to the electrodes was increased, so long as excessive sparking was not encountered. It became increasingly evident that rectifier ratings would have to be increased materially and this is being done.

At present, specifications call for voltmeters and ammeters on both the a-c and d-c sides of the rectifiers, as well as the spark rate meter.

Precipitator Alignment

To obtain the maximum power input to the precipitator, not only must the plates and wires be clean, but the plates must be spaced accurately and uniformly, and the wires must be centered. Unless these latter principals are rigidly adhered to in construction, it will not be possible to impress the optimum voltage on the electrodes.

Once the proper alignment has been achieved it is very important that nothing be done to disturb it. This

should be especially impressed upon those who are required to clean the precipitators, since heavy jarring to dislodge dust accumulations can also cause serious misalignment.

Concluding Statement

Any one of the design items mentioned could impair the collection efficiency or operation of the precipitator, if it was not properly engineered and constructed. Present experience indicates power to the electrodes to be first on the list. If sufficient power is available with automatic voltage control, fairly good performance can be obtained, even though some of the other items are not up to par.

Rapping should probably be next in importance, for without efficient cleaning of wire and plate electrodes, optimum power cannot be applied.

Gas distribution is probably third in importance; for with poor distribution, high velocity stratas will be experienced with dust re-entrainment resulting.

Gas sneakage could well be fourth, since a relatively small amount of sneakage could bypass the collection area and re-entrain some of the collected dust. This item could be large enough to ruin an otherwise fine precipitator.

In conclusion it should be borne in mind that in spite of the general opinion that a precipitator is a crude machine; it must be precisely engineered, accurately manufactured, and painstakingly erected, with careful attention given to all details, to produce an efficient and satisfactory collector.

Centrifugal Compressors Applied to Soot Blowers

J. W. LOCKE **R. G. THESING**

Diamomd Power Specialty Corp.

Centrifugal compressors are accepted and widely used in the oil, gas, chemical, steel and utility industries, the American Power Conference at Chicago, Ill., March 21–23, 1961 was recently told. The machine uses one of the simplest principles known for the compression of gases. Unlike the reciprocating compressor, which operates on the positive displacement principle, the centrifugal compressor operates on the continuous flow principle. This, in essence, accounts for its unparalleled reliability, low maintenance costs and minimum foundation requirements. The low maintenance requirement together with technological advancements in the area of operating efficiency have brought operating costs in line with reciprocating compressors. The high reliability factor eliminates the need for a stand-by compressor, which reduces installation costs and space requirements. The oil free air is advantageous not only from the standpoint of safety, but as instrumentation air.

These factors, together with the trend of increasing capacity requirements due to larger steam generator units as well as multiple boiler installations, have led to the economic feasibility of the inherently high capacity centrifugal.

In the case of working out applications for soot-blowing cycles, it must be remembered that the operating characteristic for this duty is basically one of constant pressure-variable capacity over the operating range. The range of stability will vary depending on the compressor design and the overall pressure ratio. This range, while important from the standpoint of operating flexibility, need not be a road block with the degree of load variation available in the sequenching of blowers. The characteristic curve and its envelope of efficient operation is equally important considering the range of flows and pressures desired at the various blower heads. The envelope of operation is dependent on the type of compressor controls used for providing the required pressures and flows at the soot blower heads. The final selection of the type of control used will be dependent upon the selected programming of the soot blowers.

A large utility, boiler, burning a slagging fuel, will require several different types of soot blowers to clean the heating surface. Among other items such as outside clearance space and economics, the application of each blower type depends upIn the fuel fired, type of boiler surface to be cleaned, physical arrangement of the tubes and the gas temperature zones.

Short retractable blowers (IR) are used to clean slag from the furnace wall. Blowing through a predetermined number of revolutions, the blower head pressure is normally set for 150 psig for air blowing.

Long retractable blowers (IK) are used to clean pendant type radiant surfaces and convection surfaces in the high temperature zones. They are also used to clean low temperature zones in convection passes were sufficient space is available outside the boiler wall. The blower travel speed and head pressure are functions of temperature zone and nature of deposit. In some high temperature zones, the flow rate for cleaning may be exceeded by the flow rate required for lance tube cooling.

Non-retractable, rotating, multi-nozzled blowers may be installed to clean convection surfaces in low temperature zones. These units, because of their nozzle arrangement, will have high flow rates at relative low pressures.

There are two basic types of air heater soot blowers, depending upon the type of air heater on the boiler installation. One is a high flow, multi-nozzled, traveling, "tee"-type element. The other is a low flow, single or double nozzle, oscillating-type unit.

These groups of blowers with their different flow rates represent, in effect, blocks of air. The programming of these blocks of air into an orderly blowing sequence must take into consideration many factors—the total mechanical operating time for all blowers on the boiler; the boiler cleaning prioroties; the number of complete cycles per day required to clean the boiler; the flow rates and head pressures of the particular blower units; the blowing time versus the operating time of individual units, and the type of air compressor selected are all major factors affecting this sequencing.

A simple way to determine....

Economic Thickness of Insulation for Flat Plates and Pipes

Described here is "a procedure that will mean millions of dollars of direct, quickly realized savings to all heat-using industry, and long-term improved efficiency for most energy consuming processes in all types of industry." The method is explained and a numerical problem and solution presented. Please note in the article the directions for obtaining the manual required for this procedure.

By WILLIAM C. TURNER

Union Carbide Chemicals Company

THE PRIME function of insulation is to limit the wasteful dissipation of heat. It makes the economical transportation of heat possible. Insulation can be compared to a leaky wall of a pipe used to transport water. The term "leaky" is used advisedly because no insulation can completely stop the escape of heat through it. All the water that leaks from a hole in a pipe is subtracted from the amount put in. The difference is the usable amount flowing from the delivery end. This phenomenon is similar to heat flow in insulated lines; the heat that is wasted cannot be included in the usable amount. The greater the thickness of proper insulation the smaller the loss of heat through the conduit walls.

Heat energy has a monetary value. However, there is no direct conversion from Btu's into dollars, as the cost of heat depends upon many factors, such as its form, location and use. For example: the Btu's in gasoline are generally more costly than those from coal. Also Btu's obtainable from coal are less expensive at the mine than at some distant point to which coal must be moved to be used. Likewise, the cost of installed insulation varies with materials used, location, labor rates and other factors. The simple statement that "the greater the thickness of insulation the smaller the loss of heat" may then be restated as "the greater the cost of insulation the smaller the cost of heat loss." Depending upon the cost of insulation and the cost of the heat, a certain definite thickness of insulation provides the lowest total cost.

This basic truth has been recognized for years. At a meeting of the American Society of Mechanical Engineers in New York City on December 6, 1926, Mr. L. B. McMillan presented a paper on "Heat Transfer Through Insulation" in which he presented the formulas for calculating the economic thickness of insulation. Mr. McMillan's work was the basis of our study.

A Complex Process is Simplified

Wherein then is this study of economic thickness new? It is new in that it presents a practical method of using basic knowledge which formerly was in a form so complex that it was seldom used. Although a detailed explanation of the McMillan formula will not be attempted here, the formula is shown to illustrate why it has been used only infrequently, because of its difficulty of solution. The McMillan equation to determine the most economical insulation thickness for cylindrical surfaces, one material, is:

$$\left(r_2 \log_e \frac{r_2}{r_1} + R_s k\right) \sqrt{\frac{2r_2 - r_1}{r_2 - R_s k}} = \sqrt{\frac{\left[\frac{Y(t_0 - t_a)M}{1{,}000{,}000}\right] k}{b}}$$

This equation is to be solved to determine r_2.

Where:

- b = cost of insulation per year including capital investment, interest, depreciation, maintenance (dollars/sq ft/yr)
- k = conductivity of insulation (Btu/sq ft/hr/deg F/in.)
- r_1 = inside radius of insulation (inches)
- r_2 = outside radius of insulation (inches)
- t_a = temperature of air (deg F)
- t_0 = operating temperature of pipe (deg F)
- M = value of heat per 1,000,000 Btu including consideration of fuel cost, capital investment, interest, depreciation, maintenance (dollars)
- R_s = is sum of resistances, including surface resistance (deg F-hr sq ft/Btu)
- Y = hours operation per year

Although the formula is complicated, the physical laws upon which it is based are quite simple and can best be understood from graphs.

To illustrate our first statement "the greater the thickness of insulation the lower the heat loss" (see Fig. 1.) Notice that on a bare pipe or piece of equipment the heat loss curve runs completely off the chart. Also notice that after insulation is installed it becomes necessary to double the insulation thickness to cut the remaining heat loss in half.

As previously stated, Btu's have a monetary value; therefore the scale showing "Btu's" can be replaced by dollars per year. This in no way affects the curve as the vertical scale can always be converted to the proper units.

Cost Factors Explained

Several factors enter into the cost of heat on an annual basis. All must be expressed in terms of the same unit of heat energy. These factors are:

1. Cost of fuel
2. Capital investment in heat producing equipment and the heat distribution system to point of use
3. Cost of money for capital investment
4. Interest on investment
5. Depreciation period
6. Maintenance
7. Number of hours of operation per year

As the left-hand scale is in $/yr., insulation costs can be plotted on the same chart. So, in proper units (such as linear feet), the cost of insulation per year per inch of thickness is also plotted. (See Fig. 2.)

Most of the cost factors of heat also apply to insulation. These are:

1. Capital investment of installed insulation
2. Cost of money for capital investment
3. Interest on investment
4. Depreciation period
5. Maintenance cost

The total cost per year then becomes cost of lost heat per year added to the insulation cost per year. This curve is shown in Fig. 3. It should be noticed that the

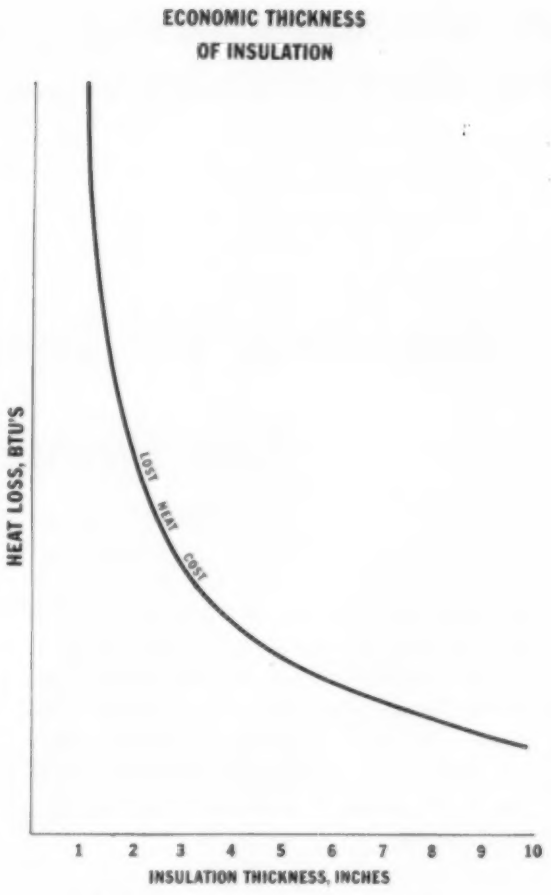


Fig. 1—Relationship of heat loss and thickness

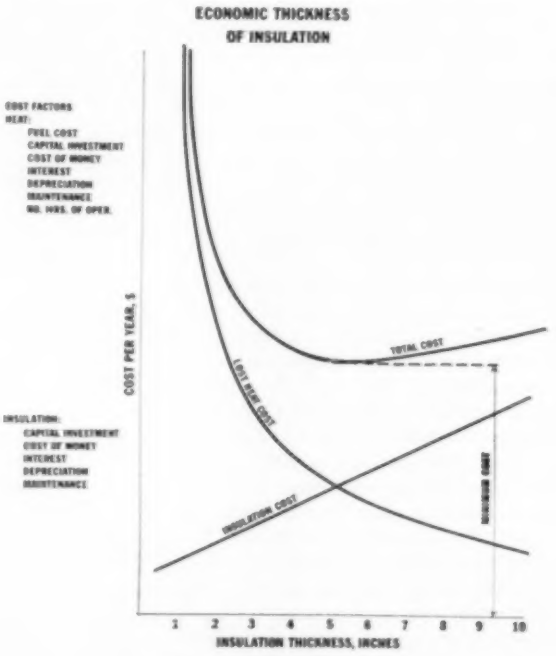


Fig. 2—Variation of costs

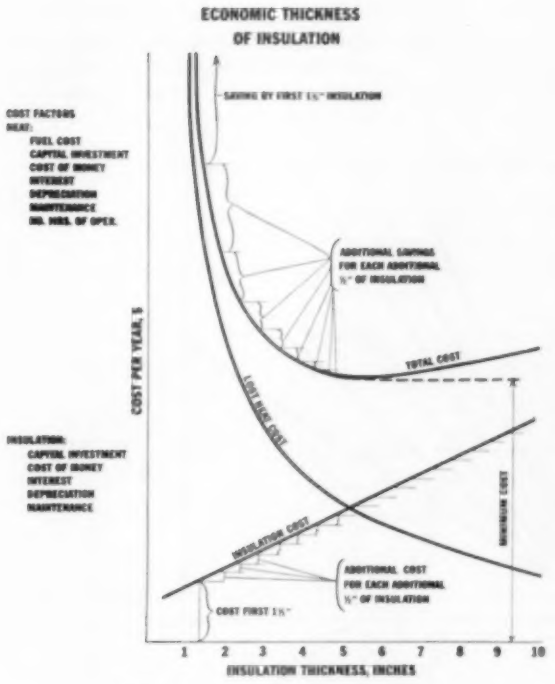


Fig. 3—Thickness-savings relationships

total cost is quite high when an insufficient insulation thickness is used; it drops to a minimum when the optimum thickness is used; then rises again when an uneconomical increased thickness of insulation is chosen. The low point of the combined curve determines the economic thickness of insulation.

These curves on insulation economics have been misinterpreted since the time they were presented by Mr. McMillan. The basic cause of this misinterpretation was the extremely high order of savings inherent in the use of any insulation. Depending upon temperature and cost factors the first inch of insulation provides a return on investment of several hundred per cent per year. Upon examination of the related costs where temperature differences and all other factors dictate thicker insulation, the last one-half inch added, though much less rewarding than the first, may still provide a return on investment of 30 per cent per year or more. These increments of investment are shown on Fig. 3.

The Problem—Too Many Variables

This graph, while typical, is only one of a family of curves. Its vertical scale of cost, its curve of heat costs, and insulation costs will be somewhat different for each combination of values. The number of different cost factors has already been mentioned. In addition to the cost factors the following physical factors must also enter into the final determination of the curves:

1. Temperature difference
2. Shape
   a. Flat
   b. Curved
      (1) Inside radius
      (2) Outside radius
3. Conductivity of insulation
   a. Mean temperature

An answer to economic thickness could be obtained by having a curve for each set of variable conditions of costs, shape, conductivity of insulation, and temperature difference. Some items of cost which affect both cost of heat and cost of insulation can be used as constants in accordance with good business practices. Many others are variable with a number of increments over a wide range. Practical limits are shown in the manual. The sum of the number of curves required would be all the single increments multiplied together or: $(7\times4\times90\times21\times4\times22\times15\times35\times22)$ which equals 53,787,880,000. Each of these charts would vary the insulation thickness in $^1/_2$ in. increments from $^1/_2$ in. to 10 in.

It is understandable that no one would start to make this number of charts.

In the calculation of economic thickness, even by machine, these myriads of figures would still be present. Some means had to be devised by which this impasse could be resolved sensibly.

A Practical Number of Variables Achieved

The particular device that broke the impasse was a separation of cost factors from the thermal constants. Mr. William Hollenbeck discovered that the cost factors could be resolved into a single variable which could determine the curves in proper relationship to the thermal equations. This was called the "D" factor.

Another observation, the use of which simplified calculations, was the fact that plotted insulation costs fell into a family of straight line curves having a common origin. The tangent angle and location of any curve on the graph may be determined by the common origin and a single point on the curve.

The application of these two observations brought the calculations required within reasonable limits.

Working on this system Union Carbide Chemical Company and West Virginia University independently derived the equations. These independently derived equations were checked and cross-checked until complete agreement was reached. Our Union Carbide Chemicals Company Accounting Department and the Professor of Economics at West Virginia University also concurred.

The problem was set up to run on an IBM 650 Computer. The program was arranged for the machine to calculate the total cost curve from data cards. For each set of conditions it started at $^1/_2$ in. insulation thickness and calculated the slope of the curve. Where insulation thickness was too thin the computer would indicate a negative number. When the total cost curve reached the lowest point and started up the other way the number would be positive. The computer was directed to record the insulation thickness where the first positive slope occurred. In this manner practical manufactured insulation thicknesses based on accepted dimensional standards of $^1/_2$ in. increments were the answers instead of meaningless decimals. In this manner all the *53,787,880,000 possible combinations of data were resolved into twenty nominal manufactured thicknesses.*

(Editor's Note: The Manual referred to below contains all the tables and curves needed to find optimum insulation

thickness by this new method. The manual will be available after June 15 from the National Insulation Manufacturers' Association, 441 Lexington Avenue, New York 17, New York. Cost of the manual will be $10.00.)

Use of Manual

In past discussions the first question usually asked was how this Manual can determine the cost of heat and cost of insulation. These are set up as variables and

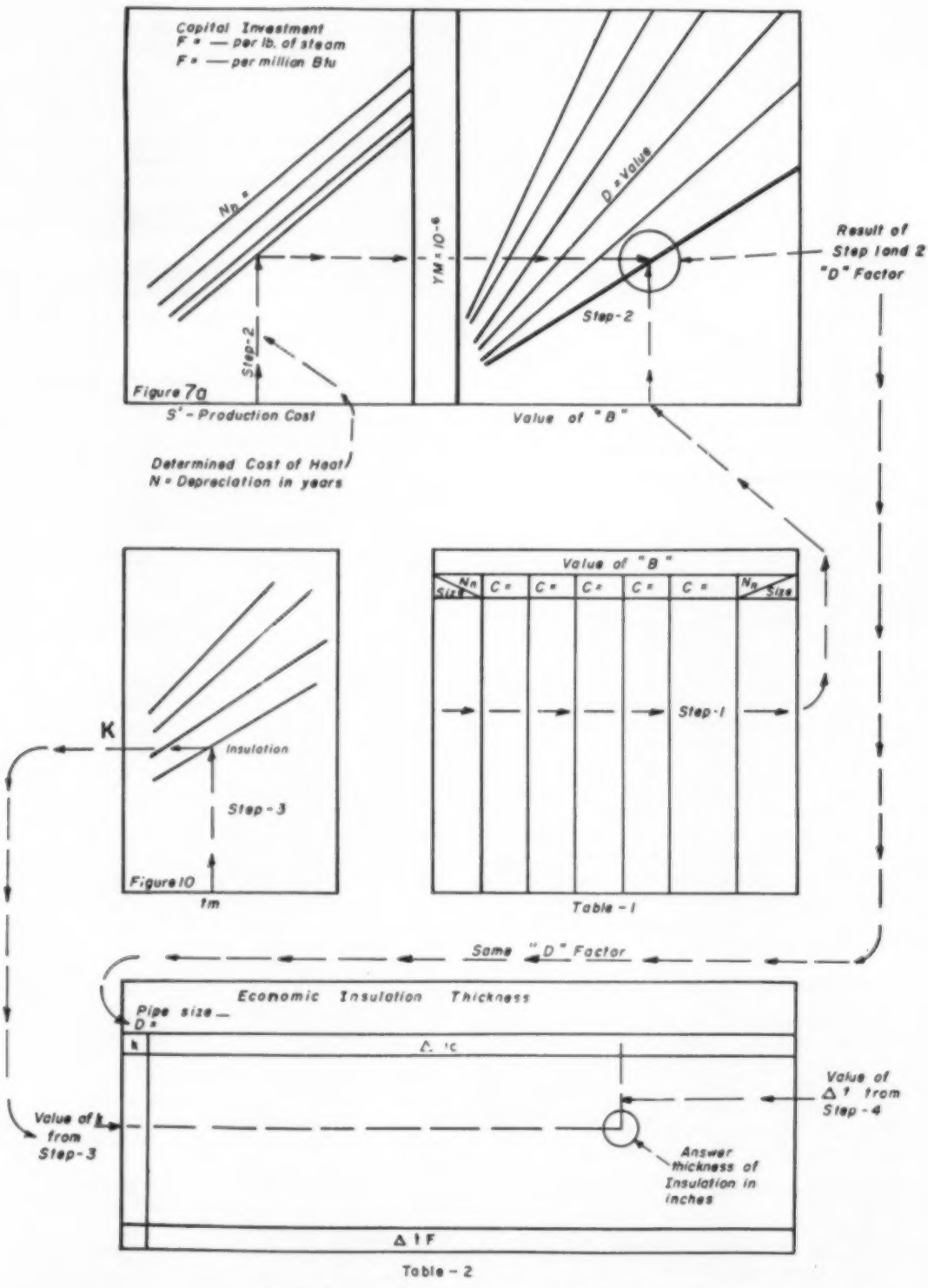


Fig. 4—Composite of charts and curves from the manual

must be determined by the user. There are various methods and accounting systems which are used for determination of cost. The relative merits of these systems are of no concern to the Manual. Capital investment and cost figures which are established and accepted by management, no matter what method is used to establish them, become the data for determination of economic thickness.

To illustrate the use of the Manual, the following data are assumed. They are only for illustration and should not be taken as the representative or recommended cost values.

Sample Problem

OPERATING DATA

Nominal pipe size	4 in.
Operating temperature (t_o)	720 F
Ambient temperature (t_{ar})	5 F
Type of insulation	High temperature calcium silicate

COST DATA

Depreciation period of insulation (N_a)	15 years
Depreciation period of plant (N_b)	20 years
Plant capital investment (F)	$13 per lb steam per hr
Total production cost of steam (S')	$1.143 per 1000 lb steam
Installed cost of one inch insulation on a one inch pipe (C)	$1.35 per (linear ft)(in.)

Solution

Step 1. Using Table 1 (see Fig. 4, Step 1), note that the given value of $C = 1.35$ is NOT tabulated. Therefore, apply the normal rules of interpolation and use the section of $C = 1.40$ (Installed cost of 1 in. insulation on a 1 in. pipe) and the 15-year column (N_a), read down to the 4 in. pipe size. At this point read the Cost of Insulation Factor B.

The value of B is 0.737.

Step 2. The Plant Capital Investment (F) is given as 13 dollars per pound of steam per hour. This value is NOT included in Manual Figures 4 to 9. Therefore, apply the normal rules of interpolation. Use Manual Figures 7a and 7b for a capital investment of 14 dollars per pound of steam per hour. Enter the chart at the total Production Cost of Steam (S') = 1.143 and read up until the curve for 20-year depreciation ($N_b = 20$) is reached, then proceed horizontally to the intersection of the vertical line through the value of B (0.737) as determined in Step 1. The point of intersection is about one-fourth the distance between the $D = 0.0120$ curve and the $D = 0.0150$ curve. Apply the normal rules of interpolation. Use the smaller value.

The value of D is 0.0120.

Step 3. Mean temperature (t_m) = [operating temperature (t_o + ambient temp. (t_{ar})]/2; $t_m = [720 + (-5)]/2 = (715/2) = 357.5$ F. Use Manual Figure 10. Enter the chart at 357.5 F, go up to the High Temperature Calcium Silicate curve and read k factor (conductivity) in Btu/hr/sq ft/in./deg F) on the left axis as 0.49. This value should be adjusted to the nearest tenth.

The value of k is 0.5.

Step 4. Temperature differential, Δt = operating temperature (t_o) − ambient temperature (t_{ar}); $\Delta t = 720 - (-5) = 725$ F. The economic insulation thickness may now be determined by the use of Table 2. Find the 4 in. pipe section, then locate the portion with a value of $D = 0.0120$. Enter the table where $k = 0.5$ and go horizontally to the column with $\Delta t = 725$ F. This value is NOT tabulated. Therefore, apply the normal rules of interpolation, use the column of $\Delta t = 738$ and read 3.5.

The economic thickness for the conditions given is 3½ inches.

By the Manual this solution has only taken a few minutes. The same problem solved by an excellent mathematician by the original McMillan formulas would have taken approximately four hours.

Liberation of Pyrite from Steam Coals

R. A. Glenn and **R. D. Harris** of Bituminous Coal Research, Inc., appearing before the American Power Conference took on the subject "Liberation of Pyrite from Steam Coals." New methods are needed for reducing the sulfur conten. of steam coals below that now attainable by conventional coal cleaning processes. Such methods are of interest to power plant operators because of the close relation of the sulfur content of steam plant fuels to thermal efficiencies, to plant maintenance costs, and to air pollution control.

At present, the methods available for the removal of sulfur dioxide from stack gases are based on wet scrubbing procedures and are economically unattractive to power plant operators. Consequently, alternative methods for controlling sulfur dioxide emissions are being considered, particularly methods for reducing the amount of sulfur in coal prior to its being burned.

Modification of the pulverization process offers a promising means of reducing the sulfur content of coal prior to its being burned. For this to be possible, however, practical methods are needed (*a*) for liberating the pyrite present in coals, and (*b*) for removing the pyrite once it has been liberated. This is believed possible if advantage is taken of the differences in properties of pyrite and of the coal substance.

The degree of sulfur removal attainable in commercial coal cleaning has been reported by the U. S. Bureau of Mines in a study of a coal from a single mine. The amount of "fixed pyrite," pyrite, which was not removable by conventional cleaning methods, represents about half the total pyrite present.

Where the pyritic sulfur is mainly associated with mineral matter, there are good prospects of its removal by adapting presently available equipment. Removal of the pyritic grains that are embedded in the organic matter will require the development of improved equipment for processing coal in the −200 mesh size.

The author reviews developments in the application and operation of the shaft driven boiler feed pump, describes some installations and ventures predictions for the future.

Fluid Drives for Turbine-Generator Driven Boiler Feed Pumps*

by RICHARD D. O'NEIL†
American-Standard

Introduction

TODAY it has become almost trite to mention the advances in steam-power generation equipment over the past ten to fifteen years. It has become quite common to discuss unit generating capacities three to six times those of fifteen years ago. It is not unusual to hear of plans or studies of unit capacities eight to twelve times greater than those of the immediate postwar period.

There have been, and there will be, advances made by revolutionary design ideas for turbines, generators, boilers and their auxiliary components. However, not all advances have been a result of revolutionary designs. Some advances have been made by taking proved, field experienced components and applying new concepts for the utilization of these components.

One such idea which has developed in the past six years is the concept of driving boiler feed pumps from the main turbine generator shaft through a fluid drive, alternately termed a hydraulic coupling.

It is the purpose of this paper to:

1. Describe and compare the main shaft fluid drive design with motor driven designs.
2. Discuss various main shaft pump and drive arrangements.
3. Report on operating experience.
4. Express some thoughts on future aspects of this main shaft concept.

Development and Design

Some six years ago during the design of the Astoria Station Unit No. 3, the Consolidated Edison of New York engineers sought a better means of driving the boiler feed pump or pumps. Consideration turned toward utilizing the main turbine-generator as the prime mover for a single full capacity feed pump. Because of its characteristics of adjustable speed, ability to clutch and de-clutch, and for protective reasons, a hydraulic coupling seemed desirable.

At about this same time, design work was proceeding on Bergen Station Unit Number 1 of Public Service Electric and Gas. Here too, consideration turned to use of the main shaft drive idea. In this case, a cross-compound, 3600/3600 rpm unit provided two generator shaft extensions for driving two half-size pumps through hydraulic couplings.

The requirements for the hydraulic couplings for these first jobs were 12,000 hp for the Astoria unit and 4500 hp for each shaft of the Bergen unit. A number of motor driven hydraulic couplings had been built equal to, or greater than, the Bergen requirements. The Astoria fluid drive, while it was then the largest ever offered for boiler feed pump service, was not the first of this capacity to be manufactured.

No revolutionary or untried design of fluid drive was necessary. This was a design problem of adaptation of motor driven designs to the specific demands and to the ultimate in reliability so necessary for the main shaft concept.

Some of the design features required by on-the-shaft drives which are not normally needed for motor driven boiler feed pumps are:

1. Removal of accessory equipment, such as pumps, coolers, filters, to an area below the drive. On motor driven units, such accessories are integral with or located immediately adjacent to the drive. To obtain a neat appearance, all piping connections are on the bottom. Accessory equipment is grouped and base mounted nor-

* Presented as "Development and Operation of Fluid Drives for Turbine-Generator Driven Boiler Feed Pumps" at the American Power Conference, Chicago, March 1961.
† Product Manager—Fluid Drives.

mally into two packages for simplicity of field installation.

2. Hydraulic coupling is side mounted with the lower portion of its housing or reservoir recessed in a well through the turbine floor. Thus coupling center line height above the floor matches that of the turbine generator.

3. Thrust bearings are designed with additional capacity to take axial thrust produced by expansion of the generator shaft. All hydraulic couplings are designed to take care of their internal thrust, but the additional thrust due to generator shaft expansion can be as much as 50 per cent of the normal or internal thrust.

4. A holding brake is installed on the hydraulic coupling output shaft. With the coupling declutched, this brake stops and holds the feed pump shaft. When the brake mechanical locks are set, the boiler feed pump may be inspected without interrupting generator output.

5. The rotors (driving, driven and rotating casing members) are, up to 23 in. effective diameter, the same on motor drive and main shaft drive units. Above 23 in. diam a high torque, high speed design previously applied on other than feed pumps is used.

These rotors, Fig. 1, are made from SAE 4340 aircraft quality steel forgings. A series of milling and heat treat operations produces the integral vane construction. The vane ring, which is a close tolerance fit to the vanes, is mechanically locked to the driving member. To minimize all points of stress concentration due to tool marks or other imperfections, the whole rotor assembly is polished to almost a mirror finish.

Hydraulic Coupling Operation

With the aforementioned differences in construction, it can be said that the principles and method of operation are the same in the main shaft hydraulic couplings as in the motor driven units.

A cross-section, Fig. 2, of a typical, 12,000 hp, main shaft hydraulic coupling shows its major components.

In operation, the independent hydraulic coupling auxiliary pumps withdraw the power transmission fluid from the base or reservoir. On lower capacity units, these pumps are integral, being gear driven from the input shaft. The fluid from the reservoir is cooled and filtered returning to the main unit for power transmission and lubrication.

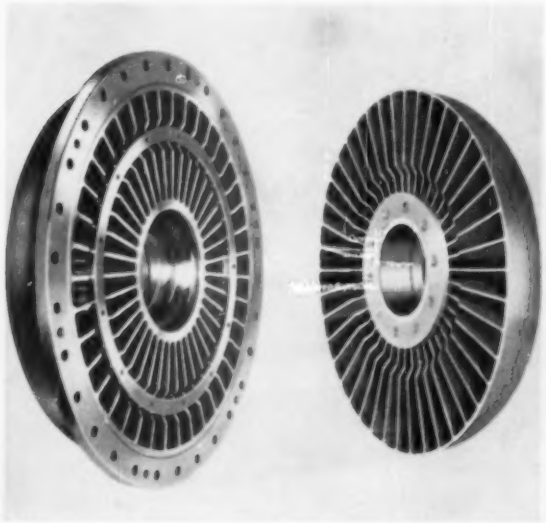


Fig. 1—Driving and driven members of a 12,000 hp fluid drive

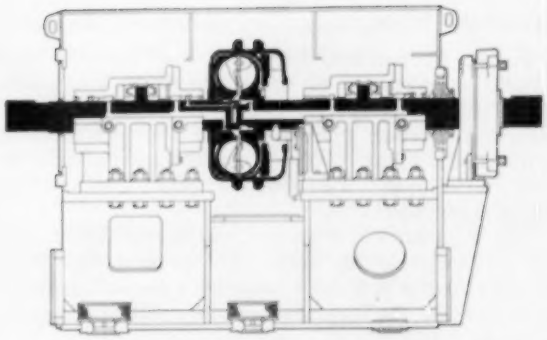


Fig. 2—Cross-section of typical 12,000 hp fluid drive

The lubrication fluid enters an inside manifold extending almost the length of the unit. Separate supply lines feed each of the sleeve radial bearings and each of the double Kingsbury thrust bearings. These supply lines are orificed to insure proper distribution.

The power transmission fluid returning from the coolers also enters an inside manifold. From the manifold this fluid is fed into both pillow blocks, enters radial, drilled ports in both shafts and then through the axial shaft passages into the rotors.

Power is transmitted from the driving member to the driven member by a fluid vortex. The driving member pumps the fluid into the driven member where the kinetic energy of the fluid is absorbed and transmitted to the output shaft. Output speed is adjusted by regulating the amount of fluid in the vortex between the driving and driven members.

The scoop tube, acting as a sliding weir, adjusts fluid level in the rotors. The result is stepless control of feed pump speed.

Pump and Drive Arrangements

At present there are 32 hydraulic couplings, in service or on order, for main shaft drive application. These are being employed on 22 generating units. Of the 22 generating units, full sized pumps have been selected for 11 units and nominal half sized pumps for 11 units.

Full Size Pump Installations

Table I lists the generating units using full size feed pumps.

TABLE I—MAIN SHAFT, FULL-SIZE, FLUID DRIVE, BOILER FEED PUMPS

User	Station	Unit Rating, Kw	Maximum Drive Hp
Con. Edison	Astoria No. 3	335,000	12,000
Con. Edison	Arthur Kill	325,000	12,000
Con. Edison	Astoria No. 4	357,000	12,000
Con. Edison	Astoria No. 5	357,000	12,000
Louisiana P. & L.	Little Gypsy	223,000	8,000
So. Cal. Edison	Hunt. Beach No. 3	210,000	10,000
So. Cal. Edison	Hunt. Beach No. 4	210,000	10,000
So. Cal. Edison	Alamitos No. 3	310,000	13,000
So. Cal. Edison	Alamitos No. 4	310,000	13,000
Con. Edison	Ravenswood No. 1	363,000	13,500
Con. Edison	Ravenswood No. 2	363,000	13,500

Consolidated Edison

The six Consolidated Edison units all have the same general arrangement of drive and boiler feed pump. Fig. 3 shows the first Astoria unit. These are cross-compound turbine generators with the high pressure turbine rotating at 3600 rpm and the low pressure turbine at 1800 rpm. The main feed pump is driven from the 3600 rpm shaft.

The fluid drives for these Consolidated Edison units are all essentially the same. The three Astoria drives and the Arthur Kill drive transmit a normal maximum of 12,000 hp; the Ravenswood drives will handle 13,500 hp each.

Lubrication requirements of the boiler feed pumps are taken care of by the fluid drive system. Cooled, filtered oil is continuously supplied to the feed pump bearings and returned to the fluid drive reservoir.

Originally the Astoria No. 3 had two half size motor driven feed pumps which could be used for start up and 100 per cent standby. Reliability of operation has permitted the removal of one of these half size spares to a subsequent generator unit.

Louisiana Power and Light

The new fully automated Little Gypsy Station of Louisiana Power and Light will employ a full size feed pump driven from the generator shaft. The hydraulic coupling will transmit a nominal maximum of 8000 hp to drive the feed pump of this single shaft, 223-Mw, outdoor type machine.

Design of the hydraulic coupling lubrication and control system is an integrated one for Little Gypsy. In addition, consideration was given to emergency lubrication needs during turbine coast-down following possible loss of alternating current source. A small direct current motor driven lubricating pump, actuated on low pressure, is provided against this emergency.

Southern California Edison

The Southern California Edison, Huntington Beach 3 and 4 and Alamitos 3 and 4 machines are also of the

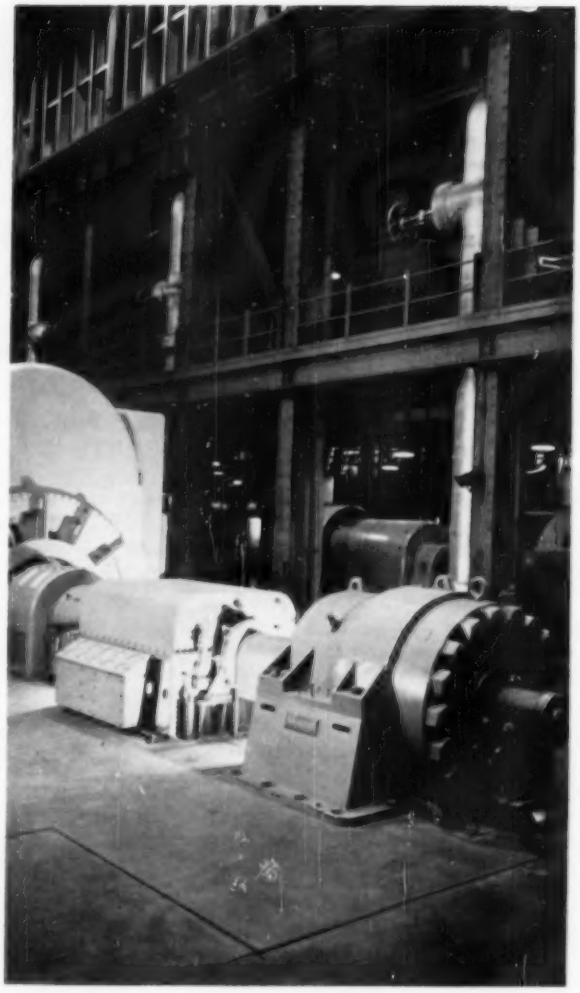


Fig. 3—View of Consolidated Edison, Astoria No. 3 drive and pump

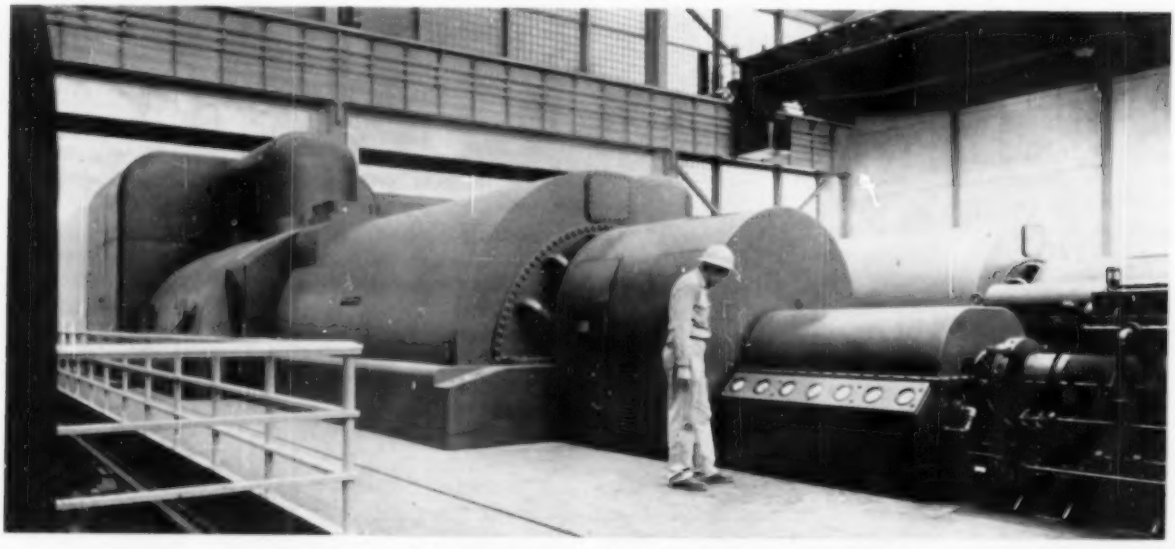


Fig. 4—View of Public Service Electric and Gas, Bergen No. 1 drive and pump

outdoor type. All will employ full size pumps driven from one shaft of these cross compound units. The pumps supply once-through, universal pressure boilers at Huntington Beach and controlled circulation boilers at Alamitos. The hydraulic couplings will provide stepless, adjustable pump speed over approximately a 2:1 range as required for these boilers.

All four hydraulic couplings are the same in general design. The Huntington Beach drives which transmit 10,000 hp have smaller size rotors and lower capacity auxiliaries than the 13,000 hp Alamitos drives.

The motor driven start-up pumps have been selected by Southern California Edison to be $^1/_3$ to $^1/_2$ the capacity of the main feed pumps.

Half-Size Pump Installations

Table II lists generating units using nominal half-size pumps, shaft driven.

TABLE II—MAIN SHAFT, HALF-SIZE, FLUID DRIVE, BOILER FEED PUMPS

User	Station	Unit Rating, Kw	Maximum Drive Hp
Pub. Service E. & G.	Bergen No. 1	290,000	4500
Pub. Service E. & G.	Bergen No. 2	290,000	4500
Pub. Service E. & G.	Mercer No. 1	320,000	5200
Pub. Service E. & G.	Mercer No. 2	320,000	5200
Pub. Service E. & G.	Sewaren No. 5	342,000	7000
Pacific Gas & Electric	Pittsburg No. 5	325,000	4500
Pacific Gas & Electric	Pittsburg No. 6	325,000	4500
Consumers Power	Campbell No. 1	265,000	4500
Consumers Power	Campbell No. 2	265,000	4500
Comm. Edison	Waukegan No. 8	325,000	5000
Pacific G. & E.	Morro Bay No. 3	325,000	4500

Public Service Electric and Gas

The ideas put forth by the Public Service Electric and Gas engineers for the Bergen Station drives served to identify and clarify requirements for subsequent main shaft drives. Fig. 4 shows one of the Bergen units.

The two 290 Mw, cross-compound 3600/3600 rpm machines have half-size pumps driven from each generator shaft. All four hydraulic couplings are of the same design and transmit, at unit rating, 4250 hp each to drive, through step-up gears, the boiler feed pumps at 8720 rpm.

Each gear and feed pump is lubricated from the fluid drive system. The bearings of the fluid drive are lubricated from the main turbine oil system. By means of an overflow line from the fluid drive reservoir, this oil is returned to the turbine sump.

Each Bergen unit employs a half-size pump, constant speed, motor driven for start up and standby purposes.

Design and arrangement of the Mercer drives and pumps follows closely that of Bergen. Each pump requires, at unit rating, 4550 hp. Differences in the pump and drive arrangement at Mercer are the use of epicyclic gears to step up pump speed to 8840 rpm rather than the single increase gear units used at Bergen; different also is the fact that at Mercer the entire generator, hydraulic coupling, gear and boiler feed pump are mounted on an outdoor deck.

The Mercer start up or spare pumps are quarter size, constant speed motor driven.

The Sewaren No. 5 will be a 342 Mw rated machine. Here too, in the pattern of Bergen and Mercer, half size, high speed pumps will be driven from each generator shaft through hydraulic couplings and step-up gears. At unit rating each boiler feed pump will require 6250 hp.

The spare motor driven pump for Sewaren will be a one-third size pump to match the start up requirements of the once-through boiler. It is interesting to note that this pump will have an adjustable speed hydraulic coupling. The decision to use adjustable speed on the spare pump was influenced by the desire to have greater pump flexibility, to avoid complications in the high pressure piping and to eliminate the need for a throttling valve.

Pacific Gas and Electric

The Pacific Gas and Electric 325 Mw turbine-generators for Pittsburgh No. 5 and No. 6 and Morro Bay No. 3 have similar drives and pumps. Each of these three is a cross-compound 3600/3600 rpm unit of the outdoor type with half-size pumps driven from each shaft.

The hydraulic couplings transmit a nominal maximum of 4500 hp for each pump.

The hydraulic couplings are designed with an integrated system for lubrication and control oil.

Essentially there are no individual start-up or spare capacity pumps for the main shaft pumps at Pittsburgh nor on the Morro Bay unit. Piping design permits the use of pumps from other boilers at these plants.

Consumers Power

Generating Units 1 and 2 of the new Campbell Station of Consumers Power will employ four half size pumps. These 265 Mw cross-compound 3600 rpm units will each have one pump driven from the h-p-l-p turbine-generator shaft and another pump driven from the i-p-l-p turbine generator shaft.

Each of the four hydraulic couplings will transmit a normal maximum of 4500 hp.

A one-sixth size motor driven pump will serve for start up operation.

Both of these Consumers Power units will have main exciters gear driven from the generator shaft extensions.

No previous mention of exciters has been made in this paper. The main shaft driven pump units previously discussed have separate motor driven exciters. From discussions with the utilities involved, it can be generally said that the main shaft drive design did not change the previously used means of driving the exciters. Either direct driven or separately driven exciters can be used where pumps are main shaft driven.

Commonwealth Edison

The Waukegan Unit No. 8 of Commonwealth Edison will have a different pump and drive arrangement than any of the previously discussed installations. Here a combination of main shaft and motor drives for the boiler feed pumps has been selected.

One half-size 3600 rpm pump will be driven from the generator shaft extension. Another half-size 3600 rpm pump will be motor driven. Hydraulic couplings for both will transmit a maximum of 5000 hp to each pump.

Operation Experience

By January 1961 there were seven turbine-generators in commercial operation with main shaft hydraulic coupling driven boiler feed pumps. These seven units

had, in January, accumulated a combined total of approximately 33,000 hr of operation. Recent reviews of the operation history of the seven generating units indicated that, since the beginning of commercial service on each unit, there has not been an unscheduled outage due to the main shaft drive concept.

The Consolidated Edison Astoria No. 3 installation is the oldest, having first turned over in September 1958. During its initial operation, difficulty was experienced with the control equipment for the two motor driven lubricating pumps on the hydraulic coupling. Each pump is capable of meeting the lubrication needs with the other pump as a spare. As originally installed the motor overload protection was set too close. On a scheduled switchover one motor tripped; the other motor had not been reset. Complete loss of lubricating oil resulted and the bearings of the hydraulic coupling were wiped.

It was determined rather early in the operation of the Astoria drive that the original drive oil coolers were not correct. Because of poor water conditions, specifications had required bi-metal tubes. Investigation of operation proved that the coefficients of heat transfer were less than anticipated. This coupled with abnormally high water temperatures and added heat load due to design features of the hydraulic coupling made cooler size inadequate. Original coolers were replaced with ones of adequate size.

The Bergen Station No. 1 unit was placed in operation April 1959 and by mid-January 1961 had accumulated some 11,800 hr of successful operation. There have been no serious problems with the drives. One of the hydraulic couplings early evidenced some excessive noise. On a planned unit shutdown, this coupling was inspected and its rotors rebalanced. Minor difficulties were found and have been corrected in the holding brakes.

The next unit to be placed in operation was that at Arthur Kill in August of 1959. During its initial running period, a valve in an oil return line from the fluid drive reservoir to the turbine sump was accidentally closed. This backed up oil in the reservoir, flooded the drive and damaged the drive rotors. No further difficulties have occurred on the coupling at this station.

Public Service Electric and Gas started initial operation of Bergen No. 2 in February 1960 and Mercer No. 1 in December 1960. The hydraulic couplings on these units are operating without any reports of difficulties.

The Pacific Gas and Electric first main shaft drive unit started in September 1960. Approximately 2800 operating hours had accumulated by the end of January. The only difficulty experienced has been on the holding brakes. Corrective action has been taken.

The 10,000 hp shaft drive at Southern California Edison's Huntington Beach Plant, Unit No. 3, became operational in December 1960. In its brief two months of operation, no difficulty has been experienced on this drive.

At least three more of the installations referred to in this paper will be operating by this summer and the end of this year will see probably a total of a dozen in commercial service.

It would be presumptuous to expect that the drives and pumps now operating, and those in the future, will all be started and operated without difficulty. It can be said that the operation experience of main shaft drives has given an indication of good reliability and a high in-service record.

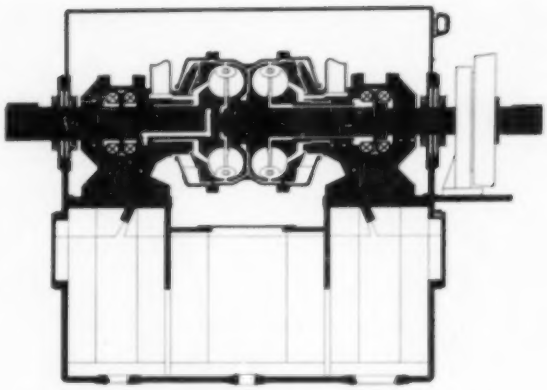


Fig. 5—Cross-section of preliminary design, 25,000 hp fluid drive

Future Developments

The main shaft pump drive concept has received increasing study and attention. Some thoughts on future developments of this idea can briefly be discussed.

The first of these ideas is hardly in the future since plans to use it are underway. The idea is to place the boiler feed pump and its drive on the turbine end rather than the generator end. The Pacific Gas and Electric, Morro Bay No. 3 will utilize such an arrangement. At least three other generating units now in the planning stage by various utilities are also considering the turbine end mounting.

Studies indicate considerable additional savings in placing the pump on the turbine shaft. Some estimates have been made that on particular installations, the turbine end arrangement could result in additional savings of $50,000 or greater as compared to generator end mounting.

The second of the future ideas is in the availability of higher capacity fluid drives. A year ago design work began on drive sizes to extend capability to 25,000 hp.

Preliminary design for these higher horsepower units is completed. Fig. 5 shows a cross section of this design. The basic theory and principles of operation are the same as all previous boiler feed pump fluid drives.

The design does utilize a tandem, dual rotor arrangement having two driving members and two driven members. Speed adjustment is obtained through the dual scoop tubes connected to a single control shaft to which the automatic control actuator is linked.

While the use of a dual rotor, dual scoop design has been, in this country, limited to special design defense purpose drives, such dual design has been successfully used on boiler feed pump drives in England and in European power plants.

Another future extension of the shaft drive concept lies in its extension to smaller generating units. Seemingly the pattern to date has been to consider shaft drives on 200 Mw units and larger. The economics of installation and operation and the desires of consultants and users will always govern. Apparently there has not been too much consideration given to this idea on smaller size generating plants. It is possible that the future

may see greater study and acceptability for lower capacity machines.

Summary

The idea of using the turbine-generator as the prime mover for the boiler feed pump has received increased study and acceptance.

It has been the purpose of this paper to describe the pump and drive arrangements which are being used and will be used. It is hoped that the discussion has shown the great flexibility for use of shaft driven pumps regardless of design of turbine-generator, size of pumps and other factors.

There have been in the past several years some excellent technical papers and articles by consulting and utility engineers dealing with the economic justification and advantages for main shaft hydraulic coupling driven boiler feed pumps.

While the economics of initial investment and operating costs for feed pumps and their drives will always be important, the final selection of these components must take into consideration, reliability of operation.

The combination of the adjustable speed hydraulic coupling and the modern boiler feed pump has a record of reliability proven over the past fourteen years on some eight hundred installed units, motor driven and shaft driven.

It is believed that the present is showing, and the future will confirm, the significant advance of main shaft driven pumps.

Preventing Furnace Explosions—Part 3

By W. L. LIVINGSTON and PHIL GRAY, JR.

Kreisinger Development Lab—Combustion Engineering, Inc.

The desirable firing process, the authors told an American Power Conference audience, of any combustion chamber (furnace) is based upon the intention to ignite and consume any flammable products entering the combustion chamber—as they enter the combustion chamber. The idea is simply to supply and burn the burner ingredients on the rate basis and at the same time minimize their accumulation.

The explosion process requires: (1) that burner ingredients are not consumed on the rate basis—that a flammable mixture is allowed to accumulate in appreciable quantities in the furnace (2) that after the accumulation becomes explosive, it is ignited.

Properly firing on the rate basis is the best method to keep a furnace free from explosion. The basic operating paths shown that can allow a chamber to explode are:

(1) Supply ignition energy to an explosive mixture.

(2) Providing an explosive mixture (when adequate ignition energy is present) by: (*a*) increasing the fuel-air ratio, to the explosive limit, in a fuel-lean and inert-lean mixture (*b*) decreasing the fuel-air ratio, to the explosive limit, in a fuel-rich and inert-lean mixture (*c*) decreasing the inert concentration, to the explosive limit, in an otherwise explosive fuel-air mixture. (*d*) combinations of *a*, *b* and *c*.

"It has often been stated," the authors went on to say, "that the best furnace safeguard available is a good operator." If an ideal operator can prevent furnace explosions, however, he must do so by responding to phenomena and taking some form of action. The results of an analysis of an ideal operator's fundamental thought process, as regards running a furnace, was shown by the authors in computer-type form. This analysis points out that an ideal operator covers each of the start up, operation, and shutdown functions with a protection system consisting of two distinct subsystems—one backing up the other.

Even an ideal operator must be able to perform a safety shutdown. Unfortunately the commonly used fuel-trip may not always provide a safe shutdown. In fact, if the furnace fire is lost because of a fuel-rich mixture (a more frequent occurrence than is generally realized), directly shutting off the fuel guarantees that explosive furnace conditions will ensue. A feasible method of obtaining a truly safe shutdown is to inert the furnace, by adding appropriate inert material, prior to taking any other course of action. Inerting the furnace can protect the fireside of a boiler tube as a safety valve can safeguard the water side. No matter how the firing system is operated, if the furnace is properly inerted no real danger will exist. It is the only single act in operating a furnace or combustion chamber that is always safe. The practical application of a furnace inerting system is presently being developed for use in a total furnace protecting system.

This intrinsic protection philosophy is the foundation for a practical furnace "safeguard" philosophy. This philosophy is translated into a compatible interconnection of subsystems which can replace the functions of, and provide the safety afforded by, the ideal operator.

This practical philosophy and its application to a boiler will be constantly improved and simplified as new measuring tools are developed. By implementing the fundamental philosophy with fewer and more reliable devices, the total system reliability will increase.

Although a substantial number of integrated devices are required to "safeguard" a boiler, few get more attention than the flame detector. Unfortunately for some installations, flame detectors have been given the blanket responsibility for protecting the furnace. When basic interlocks are replaced by a flame detection system, the unit is not as safe as it would be if no protection system at all were used.

Currently available flame detectors are basically yes-no devices. Although some may be suitable as operator aids, their present use in a supervisory furnace protecting subsystem is to:

1. Actuate something when the device senses that there is fire within its jurisdictional area (yes).

2. Actuate something when the device senses that there is not a fire within its jurisdictional area (no).

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The valve illustrated at the left is the Hancock Type 7130 Steel Globe Valve widely used by the power and process industries.

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American Power Conference in Review—II

LAST month's issue of COMBUSTION carried a fair portion of the American Power Conference papers held to be of value to the readers of this magazine. There were, however, so many that in the interest of space the meeting report was divided into two parts and the majority was held over for this issue.

Breed Station

A very well attended, although somewhat lengthy session, that was built around "The Breed Plant, The 500-Mw Foundation for Large, Modern, High Efficiency Unit Development" was opened by **Philip Sporn** and **S. N. Fiala,** American Electric Power Company, with their paper "The Project, The Prototype, A Look into Future Size Development." Breed is in effect a mine-mount plant. Coal for its cyclone-fired boiler is brought into the plant, a distance of 10 miles, over a captive railroad from the Thunderbird Mine of Ayrehsire Collieries. The entire output of the plant is stepped up in voltage by two 275-Mva 3-phase transformers and delivered over a 186-mile, double-circuit, 345-kv transmission line into the American Electric Power System at Olive Substation of Indiana & Michigan Electric Co.

In exploiting the supercritical pressure cycle at 3500 psi and double reheat, Breed has stepped back in pressure and temperature from far-reaching levels to attain a new plateau of thermal efficiency. This has been done under terminal conditions carefully evaluated to optimize capital and operating costs and produce energy most economically.

Since the service area of Indiana & Michigan Electric Co. has neither sources of fuel nor well-endowed plant sites, it was necessary to go outside the operating area to find water and fuel: Most significantly, in a finding on this question raised before it, the Securities and Exchange Commission clearly established both the right and the responsibility of a utility system to seek and use locations for its plants, regardless of operating area, that are necessary to provide optimum commercial energy supplies.

This idea has been in process of development on the American Electric Power System over a period of 40 years. It started in 1916 with the Windsor Plant delivering power to Canton over a 55-mile, 130-kv transmission line. It was continued with Philo and its 73-mile, 138-kv to Canton, and it was further developed with Tanners Creek. But it remained for Breed to combine use of 500,000-kw generating units with 345-kv transmission over a single tower line to bring power from an optimum plant site into an operating area almost 200 miles away.

The decision to use electric transportation for this purpose was based on these considerations:

1) Solid confidence in the reliability of 345 kv as a bulk transmission voltage.

2) Favorable economics of electric transmission vs. rail haulage even on the first unit installation.

3) Certain knowledge that electric transmission economics would improve with Unit No. 2 and particularly Unit No. 3.

4) Conviction that such a decision would be further endorsed with the first increase in freight rates. Such an increase was not long in coming.

Size extrapolation is particularly important. Experience over a long period has demonstrated that extrapolation, to be economically successful and attainable, must be quite conservative. The ratio generally should not be greater than 2 : 1. A convincing example of this practice is the 77,500-kw, 2500-psi, 940/900 F single-boiler, single-turbine generating unit placed in operation in 1941 at Twin Branch Plant. Experience with this unit led to the postwar installation of seven similar units, each designed for 150,000-kw, 2000-psi, 1050/1000-F reheat operation. All austenitic material was used on six of these machines in the high temperature parts of the turbine, including turbine piping and valves. In the last of this series, less expensive ferritic materials were substituted to check the suitability of this type of construction for 1050-F high-pressure steam service. Results were better than expected.

Successful design, manufacturing and full-scale test of ferritic materials at 1050-F service led to another projection in size, this time to 215,000-kw net output. It brought a further improvement in efficiency due to the increase in size and reheat temperature from 1000 to 1050 F. This led to a long line of excellent production units which in later versions were expanded to 225,000 kw, making the next step more difficult.

It was obvious that an increase in unit size alone would not achieve the necessary control over rising equipment and construction costs. Substantial improvement in thermal efficiency had to accompany the next size projection. The authors reported facing this particularly difficult problem of introducing new ideas into the designs of natural-circulation boilers which had sufficient promise to insure progressive improvements in larger sizes. Much larger and thermally inflexible steam drums, with equally undesirable heavy-walled pressure parts, made the once-through type of design necessary and load particularly urgent because of the rapid system growth.

The Breed Plant is the first commercial unit based on

Philo No. 6 knowledge, design and operating experience. Notwithstanding some delays and unexpected and painful troubles, it is now performing better than expected. The Breed project was announced in 1956 as the first of two 450-Mw supercritical units to be installed on the AEP System. The second unit was later scheduled as an addition to the Philip Sporn Plant. This unit, too, has now gone into commercial operation. Now rated at 500 Mw, the turbine-generators are identical except that the generator stators on the Sporn machines are water-cooled, while the Breed stators are oil-cooled. Water-cooling, so far proved entirely successful on the Sporn Unit, should add materially to compactness of future high-capacity stators.

Although operation of the supercritical unit at Philo demonstrated the feasibility of 1150 F as a suitable mate for 4500 psi, industry has been unsuccessful in producing low-enough-cost high-temperature materials to justify their cost from the resulting thermal gains. The steam conditions on the Breed Unit, therefore, were set for what then appeared to be the most attractive combination of 3500-psi, 1050-F high-pressure steam, with two stages of reheat, both to 1050 F. Our work since 1956 has not only confirmed this but has shown further economic gain in dropping initial and first-reheat temperatures 50 and 25 deg respectively.

The first paper on station design, "Steam Generator", was presented by **G. W. Bice, P. H. Koch** and **E. M. Griffin.** The steam generator for the Breed 500-Mw Unit produces 3,250,000 lb of steam per hour at 3500 psi and 1060 F temperature and has two stages of reheat, each to 1050 F. The authors held that there were two fundamental differences between the conventional 2400-psi, 1050-F/1050-F single-reheat, drum-type steam generator and the 3500-psi, 1050-F/1050-F double-reheat steam generator which had to be considered carefully. These were:

1) A unit designed to operate at 3500 psi must be of the once-through forced-flow type because at supercritical pressure no boiling occurs and there is insufficient density differential to obtain satisfactory natural circulation.

2) A unit designed for 2400 psi must provide for absorption of about one-half the total heat input for generating steam and one-half for superheating and reheating. A 3500-psi unit must provide for absorption of about one-third the total heat input for generating steam and two-thirds for superheating and reheating. This heat distribution necessitates the use of gas recirculation to avoid placing superheater surface in gas temperatures of 2700 F or higher.

Once these two fundamental differences are resolved, many of the design features that have been tried and proved in drum-type units can be incorporated in a supercritical unit.

The boiler is approximately 250 ft in height from the slag tank to the top of the air preheaters. The upper furnace and convection section are 52 ft wide by 42 ft deep and are supported directly from top steel. The lower furnace and the cyclone furnaces are bottom supported from fluid-heated support tubes which in turn hang from the top steel.

Crushed coal is fired in eight 10-ft diam cyclone furnaces which are arranged in a single row in both the front and rear walls of the fully-studded lower furnace. Recirculating gas is introduced at the top of the lower furnace, which is only 18 ft deep, and, after mixing with the furnace gas, flows through the upper furnace which is enlarged in depth to 42 ft. The gas then flows up through a convection heat-absorbing section consisting of the secondary superheater, the first and second stage primary superheater, the high-pressure reheater, the low-pressure reheater and the economizer.

Three vertical-shaft regenerative air preheaters are located above the convection section. The unit is located outdoors and is pressure-fired with no i.d. fan.

The flow of steam and water through the unit takes this pattern. Feedwater from the high-pressure heaters flows up through the economizer and then through outside downcomers to the cyclone furnaces. After passing through the cyclone furnaces, the water flows through the lower furnace and then to the upper furnace where the transition from water to steam takes place. The steam then goes through the superheater screen, the primary superheater, and finally the secondary superheater.

Furnace and burner geometry have a pronounced effect on heat absorption in both the furnace and the convection heat absorbing surfaces of a steam generator. Since heat absorption has a large influence on metal temperatures, it is important that the furnace and burner geometry result in a good distribution of heat throughout.

The relatively low ash fusion temperature of the fuel, coupled with the small percentage of the total heat that must be absorbed in the furnace, led to the use of opposed cyclone furnace firing and gas recirculation. The relatively small, fully-studded, lower furnace facilitates wet slag removal over a wide operating range and, further, aids in thorough mixing of the recirculating gas with the products of combustion. Recirculated gas—about 30 per cent at full load—is introduced at the top of the lower furnace to temper the combustion gases below the slagging temperature before they enter the convection passes and to assure an even heat input to the large upper furnace and the convection surfaces.

One characteristic of forced flow through multiple parallel tubes heated in a furnace is fundamentally different from that in a natural circulation system. With forced flow, the tubes receiving higher heat input have proportionally less flow than those receiving less heat, whereas with natural circulation, those tubes receiving more heat have the higher flow rate. Therefore, with forced flow, it is imperative that both fluid and heat distribution in the furnace be carefully balanced.

At Breed, water flows through the cyclone furnaces in a combination of series and parallel flow paths. This flow pattern is such that the fluid temperatures leaving all the cyclone furnaces, regardless of the number in operation, are balanced and close to the temperature of the furnace walls to which the cyclones are attached. Entering the bottom of the lower furnace walls, the water then flows in an up-down-up path to the lower furnace outlet where all the fluid is completely mixed. It then enters the upper furnace and flows through a similar path. After leaving the upper furnace and convection enclosure, it is given a second complete mix before entering the screen and the convection surfaces. The transition from water to steam occurs in the upper furnace walls.

During start-up of a drum-type boiler, the heat absorbed in the furnace walls starts and maintains water circulation. The only limit on firing rate is that of the gas temperature entering the superheater. A once-through forced-flow steam generator depends upon the feed pump to provide the required minimum water flow through the circuits exposed to high gas temperatures. Thus, a by-pass system to handle this start-up flow is essential. While circulating the start-up flow, the by-pass system has the following functions:

1) Reducing the pressure from above-critical in the steam generator to below-critical in the flash tank.
2) Providing a source of steam for the turbine seals.
3) Providing a source of steam for the deaerator.
4) Utilizing a large portion of the heat in the fluid entering the by-pass system to heat the feedwater in the high-pressure heaters.

The turbine by-pass valves on the Philo prototype were designed for one-third of full load flow at 950 F. To reduce both the start-up time and the initial investment, the Breed unit was designed to start up with only 10 per cent of full load flow through the turbine by-pass, with the remainder of the start-up flow through the lower temperature, less expensive, superheater by-pass valves.

During normal cold start up, a water flow of 1,100,000 lb per hour is established through the turbine by-pass valves and as the temperature of the fluid leaving the secondary superheater reaches 500 F, turbine seals and the deaerator are put into service, using steam from the flash tank. At 600 F, steam from the flash tank is available for heating the feedwater. Above 650 F the specific volume of the fluid is such that the turbine by-pass valves will no longer handle the 1,100,000 lb per hour flow and the superheater by-pass valves begin to open. At 850 F the flow through the turbine by-pass valves has been decreased to approximately 350,000 lb per hour and steam conditions are satisfactory for rolling the turbine. The use of flash tank steam to heat the feedwater greatly reduces the time and the heat input required for start up.

Between February 14 and June 21, 1960, when the unit was first paralleled, the following modifications were found necessary. Both the lower furnace and the cyclones are supported from top steel by eight "U" loops, 160 ft high, located outside the furnace walls. The support loops must be maintained at a temperature close to the average enclosure temperature to minimize differential expansion between the upper and lower furnace walls. This is done by circulating a portion of the fluid from the upper furnace outlet through the support loops. In the original design the inlet and outlet of each loop was connected upstream and downstream of orifices in the furnace outlet piping. Although the orifices were correctly sized for the load range, they were not satisfactory for transient start-up conditions. The low specific volume of the fluid passing through the orifices during start up resulted in a very low flow through the support loops with much of the heat in the fluid transferred to the heavy-wall support tubes. Consequently the fluid density difference between the down-flow and up-flow legs of the support tubes was great enough to overcome the orifice pressure drop and stop flow through the support tubes. Circulation in the support tubes was changed to down-flow in both legs by installing a pipe from the bottom of each support loop to a point further downstream in the furnace outlet piping, and relocating the orifice downstream of both legs, thus making both down-flow. This modification required less pressure drop than the alternate of merely reducing the size of the original orifice.

Soon after raising superheater outlet temperature to 850 F for the first time, a tube failure occurred in the down-flow, second section of the primary superheater. Because this tube failure occurred near the top of the superheater, fluid recirculation was suspected. Additional thermocouples were installed and data obtained showed that recirculation was taking place during transient start-up conditions. The recirculation was due to small fluid temperature unbalances which resulted in density differentials large enough to overcome the relatively low resistance to flow through this down-flow section. An examination of the temperature-enthalpy and specific volume-temperature relationships of water and steam at 3500 psi reveals the large quantity of heat required to raise temperature a few degrees in the 700 F zone, and the correspondingly high increase in specific volume (reciprocal of density) which takes place with these few degrees of temperature change.

The second section of the primary superheater was changed to up-flow to assure a positive flow through each section during all phases of operation.

When the superheater outlet temperature was raised to the transition temperature of 710—730 F, steam temperatures in the four main steam leads varied as much as 200 F. The spread in temperatures occurred each time the steam temperature was raised to the transition range, and was due to flow stagnation in one or more of the steam leads. This stagnation occurred because small fluid temperature unbalances resulted in density differentials sufficiently large to overcome the relatively low resistance to flow through the down-flow steam leads.

The secondary superheater outlet steam piping was modified to provide pressure equalization and a partial mix of the fluid. This assured flow through all sections of the secondary superheater during all start-up transient conditions and thereby prevented excessive spread in the steam lead temperatures. The maximum spread of steam lead temperatures during start ups after the alteration has been about 25 F.

Although major difficulties were encountered during start up of this unit because start-up transients were not fully anticipated, all of these difficulties have been satisfactorily corrected and performance of the unit has met or exceeded expectations. Lower than expected fluid pressure drop, along with well balanced absorption in the furnace and the convection sections, have made it possible to achieve a continuous steam output of 3,250,000 lb per hour.

C. P. Lugrin, P. G. Ipsen, J. A. Massingill and **J. A. Oliver** combined to discuss the "Turbine Generator." The Breed arrangement of turbine casings has a number of attractive features, Each element has three casings with the high-pressure section at the head end of the primary element and the first reheat section at the head end of the secondary element. The remaining two casings of each element contain the second reheat section and the triple flow low-pressure action. The division of output between the two 3600-rpm turbine elements is essentially equal at all loads, thus making possible the use of duplicate generators, exciters, buswork and transformers.

The second-reheat and low-pressure sections also are duplicates. This duplication reduces the number of spare parts, which is particularly important for this installation where spare rotors were purchased.

The location of the high-pressure and first-reheat sections on different shafts makes for a relatively uncongested layout for the high-temperature piping. There are no crossover pipes between the two elements. Therefore, no restriction is placed on keeping the primary and secondary elements close together. Furthermore, the layout problem associated with large crossover pipes between the elements is eliminated. The only crossovers go from the exhausts of the second-reheat sections to the inlets of the double-flow portions of the low-pressure sections, and these are quite short. This arrangement, coupled with the location of intercept valves right on the first- and second-reheat shells, results in small entrained steam volumes and keeps the overspeed, on loss of load, to a minimum despite the fact that both the turbine and the generator rotors have a low moment of inertia relative to their output. From this review of the advantages, it can be seen that the Breed turbine arrangement is well suited for the double-reheat cycle.

The combination of large volume flow and steam conditions of 3500 psi, 1050 F, made it desirable to use four main stop valves so that the wall thickness of the individual valve bodies will allow reasonable rates of temperature change without excessive thermal stress. The valves themselves are arranged somewhat differently than is the usual practice in that the inlet strainer is located above the valve disk on each valve instead of surrounding it. This design reduces the inside diameter of the valve body, and still further reduces the required wall thickness. The four stop valves are located in front of the turbine, and the above-seat chambers are connected together by equalizer pipes of about the same size as the main steam inlets. The equalizer pipe between the two inner valves actually serves as a chest for the four turbine by-pass valves, as well as for a start-up valve which by-passes the four main stop valves during starting.

The turbine by-pass valves are used to provide the necessary flow of water or steam through the boiler during start up. They are sized to pass about 240,000 lb per hour of 3500 psi 950 F steam and will pass about 1,000,000 lb per hour of 3500 psi, 600 F water. The location of these by-pass valves is such that they serve as the drain for the main steam piping and above-seat chambers of the main stop valves, and yet they do not cool off and fill with water when they are closed and the turbine running. Thus they stay hot and can be opened immediately after a shutdown without being subjected to rapid temperature changes.

The four main stop valves feed steam to two control-valve chests which are located on each side of the high-pressure section near the turbine floor. There are four control valves in each chest. Two servomotors located in the base under the turbine front standard operate the control valves through cam-actuated mechanisms. The eight valves feed the four main steam connections on the high-pressure section and are piped so that each high-pressure inlet is fed by one valve in each chest. This arrangement makes possible the full closed testing of all the valves in one chest while the turbine is carrying load. Eight valves were chosen rather than four to permit the use of simple unbalanced valves of a size and type which has been used widely with good success at 2400 psi.

The inner shell joint in the region of the first-stage seals against a maximum pressure difference of about 1600 psi between the first stage and the exhaust at a temperature level of about 975 F. The design to accomplish this end utilizes bolt cooling. A small amount of steam extracted from the fifth stage cools the bolts enough to increase their relaxation strength to a point where the joint will stay tight for normal periods between retightening. The outer shell joint is of conventional construction of suitable proportions, and bolt material, to seal against the exhaust pressure of about 1035 psi.

After the steam is reheated to 1050 F in the first reheater it comes back to the first-reheat section through two parallel reheat stop valves and two intercept valves. The reheat stop valves are located in front of the first-reheat section, and are the swing-gate type having a very low-pressure drop. The intercept valves are angle-body type, and bolt to the upper half of the high-pressure shell. They are operated by servomotors located in a base under the turbine front standard.

The authors continued to give detailed data on the other sections of the turbine generator. The time to bring the turbine up to speed was set at 10–15 min for either cold or hot starts instead of using longer times for cold starts. Steam conditions at start up are 3500 psi, 850 F. The turbine is brought up to speed using the stop valve by-pass with the control valves wide open. After full speed is reached pressure is built up in the control-valve chests using temperature difference across the valve chest wall as a guide. The allowable differential varies from 260 to 160 F depending on metal temperature, with the higher value permitted at temperature below 400 F. Under these limitations, time for pressurizing the steam chest varies from approximately one hour for a cold start to just a minute or less for a hot start.

After paralleling the unit, load is raised to hold the first stage shell wall temperature differential within 150 deg F. At 100-Mw load, control is transferred from the sychronizing device to the pressure governor. The turbine control valves maintain throttle pressure, and load control is transferred to the boiler feed pump and further increases are made with either the motor-operated or turbine-driven boiler feed pump.

Load is increased as determined by the proper heating rate, to approximately 180 Mw, using the motor-driven boiler feed pump. At this point, the turbine-driven pump is started and the flows from the two pumps are paralleled. The motor-driven pump is shut down as load on the turbine-driven pump is increased. The motor-driven pump can carry the unit to about 208 Mw, while the turbine-driven pump can carry the unit down to about 150 Mw.

The subject of the "Thermal Cycle Equipment and Performance" was given by **T. T. Frankenberg** and **J. A. Tillinghast**. The thermal cycle selected for Breed Plant was greatly influenced by experience with the prototype unit, Philo No. 6. Pressure for the prototype had been chosen as 4500 psi in order to extend the exploration

well into the supercritical range, and also to avoid any initial difficulties that might occur in the event that unforeseen discontinuities were encountered immediately adjacent to the critical point. By the time that Breed pressure had to be confirmed, the prototype unit at Philo had been operated successfully at both 4500 and 3500 psi, and it had been found desirable to make all starts and particularly "hot restarts" at the lower pressure. This was due only to the greater ease in matching turbine metal temperatures, since there was less expansive cooling of the steam when throttling from 3500 psi, and not to any mechanical difficulty in controlling the higher pressure.

With the pressure for Breed established at 3500 psi, it was possible to avoid the use of austenitic piping if the temperature was not above 1050 F. The history of cracking in austenitic steam lines, the economic problems, and a reluctance to base major units on a scarce and at times critical material were all factors in the decision to establish initial temperature at 1050 F. It was interesting to note that with Breed's conditions of 3500 psi and 1050 F the volume flow to the first stage nozzles was about five times greater than at Philo. This increase in volume flow permitted the internal efficiency of the high pressure turbine to be improved appreciably.

Double reheating was retained, as in the prototype, since it had caused no operating complications at Philo. For a unit in the size range of 500 Mw, double reheat provides approximately the same thermal advantage as could be achieved if both initial and reheat temperatures were raised by 80 F on a single reheat 3500- psi turbine. It was felt that over the life of the unit double reheat would deliver its incremental improvement in heat rate, with markedly fewer complications and less metallurgical uncertainty than the alternate course.

Reheat pressure-level studies showed that the highest efficiency occurred when the entropy increases across the high-pressure and first-reheat turbine stages were about equal. The change in efficiency, as pressure is varied, is not sharp. Therefore the two reheat-pressures were set at 1000 psi and 330 psi for mechanical considerations. The most potent was the volume flow through the second reheat system and the first turbine stage after reheat. By setting initial reheat pressure at the relatively high level of 1000 psi, a heater extraction point suitable for a feedwater temperature of 545 F was available, and no extraction was taken from the high-pressure turbine. To moderate the loss due to use of highly-superheated steam in feedwater heaters, the heaters next below the two reheat points were depressed in pressure so that the temperature increase in each of the two heaters on the cold reheat points was about 1.5 to 1.75 times the average rise in all heaters.

The reduction in initial pressure to 3500 psi and the knowledge, from the outset, that feedwater temperature would be in the general order of 545 F resulted in a modification of the boiler feed pump arrangement when compared with the prototype. At Philo it was found that feed-water heaters designed for service at a water pressure of 5500 psi were too costly so that it became prudent to divide the feed pumps to allow operation of the heaters at an intermediate pressure of approximately 2200 psi. At Breed the lower initial pressure improved the outlook for full pressure heaters, while the higher feedwater temperature indicated that there might be extra problems in sealing the high pressure pump if it were placed above the last heater. It was noted that a significant increase in feed pump power would be required in pumping at temperatures above 500 F. An added factor operating to change the pattern away from the prototype's was the successful operation of large, turbine driven, single boiler feed pumps on the Glen Lyn 6 series of 225-Mw units, beginning in the spring of 1957.

Turbine drive was favored because the feed pump of a supercritical unit is by far the largest single auxiliary, and numerous electrical problems would be eliminated by this choice. On once-through units a higher proportion of the feed pump head is used in overcoming system resistance, thus creating a further advantage for variable speed drive. Weighing these factors it was determined to use split pump arrangement with the main pump turbine driven, while three half-size motor driven low pressure feed pumps were used at the outlet of the deaerator, pumping through two intermediate pressure heaters at about 600 psi before entering the suction of the main boiler feed pump. By this method only heaters 8 and 9 are designed for 4500 psi tube pressure, and the major portion of pump power is introduced at a water temperature of 440 F. Heaters 8 and 9 were purchased at twin stream, since they were not only less costly when all valves and controls were considered, but would provide the added advantage of permitting full flow at nearly full temperature through either stream. Heaters 1 through 4 were arranged as a single stream with provision for bypassing any one individually on the water side. Following an AEP practice of 10 or 12 years standing, the two lowest pressure heaters were not provided with steam valves, either shut-off or check valves. Terminal temperature differences were carefully evaluated on all heaters.

The main feed pump is rated at 7200 gpm at a head of 11,170 ft while operating at 3600 rpm. The driver is a 20,000-hp eleven-stage impulse turbine exhausting to its own 18,000-sq ft condenser. Experience with higher speed turbines and pumps was exceedingly good. However, for this unit the relatively low speed of 3600 rpm was chosen to avoid the expense of double-flow exhaust stages. Because of the very large flow through the feed pump, this speed did not represent a serious compromise with regard to the desirable specific speed requirements.

Operation of this turbine and pump has been most satisfactory with no outage of the turbine traceable to this equipment. Pump efficiency has been about 3 per cent better than its high guarantee. However, an unexpected and unsatisfactory surge condition has been experienced at low flow conditions. This is assessed as a function of the low attack angle between water and impeller, accentuated by a high ratio of eye-to-impeller diameters. Model studies are expected to resolve the basic cause of this phenomenon, the only unforseen result of a large extrapolation in pump design.

A standard requirement of supercritical units, at least at that time, was the provision that one-third full flow must be passing through the steam generator before any firing of the unit was permitted. This precluded the opportunity for a boot-strap type of operation with a small start-up pump, which had proved a highly satisfactory method of bringing the Glen Lyn 6 series from about 400 to 500 psi (where the start-up pump becomes inadequate) to full flow and a pressure of 2000 psi.

The requirement of one-third flow against full pressure made for a very large start-up pump. It was determined that the incremental cost of increasing its size from one-third to one-half flow was fully justified by the additional back up it would provide for the main feed pump. This provided an operating advantage in the added overlap between the load at which the main pump drops out and the upper limit at which the motor-driven pump can safely handle the load. The temporary expedient to overcome the low flow surging consisted of a substantial increase in emergency leak-off flow from both pumps and it was fortunate that some added overlap had been provided for the critical step of transferring from one pump to the other.

Variable speed was chosen for the start-up pump also, to provide for gradual pressure build up on the boiler, thus avoiding unnecessary pressure breakdown at the turbine by-pass valves when handling cold water. Variable speed was achieved by driving the pump from a 13,500 hp wound-rotor motor with a speed range of approximately 360 to 720 rpm, and then through a speed-increasing gear to a top speed of 3500 rpm. The motor-driven feed pump is a 9-stage barrel- ype pump capable of delivering 3600 gpm against a total dynamic head of 11,170 ft.

A. S. Grimes and **W. S. Morgan** collaborated on the paper "Instrumentation and Controls." The instrumentation and control for Breed Unit 1 extends the centralized control philosophy developed on the AEP system in the past 20 years. The unit is provided with a central control room in which are located all of the instruments and controls necessary to start up, operate and shut down the unit as well as the controls for coal handling, the transmission switchyard, and Dequine substation located some 90 miles away. In addition, the control room houses the instrumentation, including an analogue computer, for monitoring the thermal performance of the unit.

The use of a power-generating cycle with a once-through type steam generator requires such close coordination of the boiler controls and the turbine-generator controls that in effect they become one control system. The primary control signal is the desired generation. This signal may be applied manually by the operator or automatically by the system load control. This primary signal sets the rate of feed water flow to the steam generator. A subcontrol loop adjusts the pump speed as required to maintain the desired rate of flow.

The control signal positions either a potentiometer in the amplidyne of the liquid rheostat for the motor-driven pump or a pneumatically operated speed changer in the governor of the pump turbine.

Steam pressure at the turbine throttle is controlled by positioning the turbine control valves. Under emergency conditions, the turbine speed governor can limit the turbine control valve opening to control overspeed and a pressure limiting control then comes into action to reduce feedwater pump speed if necessary to hold the throttle pressure at a safe maximum value.

Firing rate is adjusted to control the main steam temperature at the desired value. Since there are four superheaters in parallel some means of maintaining equal outlet steam temperatures is required. This is done by spraying a fixed quantity of water into the inlet of one superheater. The outlet temperature of this superheater is then used as the control signal for the combustion control. The other three superheaters are equipped with conventional spray attemperators to control their outlet temperatures to a value equal to the outlet temperature of the control superheater.

In conventional terminology, the combustion control loop is of the two-element type. That is, it uses feedwater flow rate and steam temperature as the two elements. The control loop acts to balance the energy input to the steam generator to the feedwater flow rate. Deviation of the steam temperature from the desired value causes a correction to be applied to the energy input. The combustion control has proportional plus reset response with a very minor amount of rate response. The proportional response is adjusted automatically to compensate for the number of cyclones in service.

The spray attemperator controls have proportional plus reset plus rate response with the rate and reset in parallel. Originally there were two water control valves for each superheater section. Changes made to the steam generator after preliminary operation led to eliminating one of the valves in each set.

The superheater section which is used as the control section for the firing rate also has a spray attemperator control for protection against abnormally high steam temperature from mis-operation of the firing controls.

The combustion air flow rate is controlled in parallel with the fuel rate with automatic correction of the overall fuel-air ratio from automatic analysis of the exit flue gas. Gas analyzers are provided at each cyclone outlet to correct the fuel-air ratios of the individual cyclones.

In the event of an accidental loss of a substantial part of the firing capacity, a low steam temperature override controller will reduce the feedwater flow rate to a value equivalent to the limited firing rate.

Three sets of gas proportioning dampers are used to control the distribution of energy among the two reheaters and the superheater. These dampers are positioned so as to maintain each of the two reheat steam temperatures at their desired value. The second reheat steam temperature must be reduced under conditions of low load and high turbine back pressure in order to prevent the turbine exhaust steam from being superheated. This is accomplished by providing a set point adjustment from the turbine exhaust steam temperature controller.

The Breed unit is equipped with two sets of by-pass valves and a pressure breakdown system to provide the required flow capacity by-passing the turbine during steam generator start up. One set of valves receives water or steam from the inlet of the turbine stop valves. These valves are operated sequentially from a hydraulically driven camshaft. The second set of eight valves are connected to the outlet of the first section of primary superheater. These valves are air-operated with their positioners adjusted so that the eight valves open as two valves in sequence.

The control system is single-element, operating to maintain the throttle steam pressure at a preset value. Either or both sets of valves can be on automatic control. When both sets of valves are in service, there is provision for changing each set's participation so that a smooth transfer of flow from one set to the other may be made. Pitot tubes in the main steam loads are used to determine the portion of the flow through the turbine by-pass valves under start-up conditions.

Other control loops maintain the water level and steam pressure in the flash tank and control the sequence of using flashed steam for deaerator pegging, turbine steam seals, and high-pressure heater pegging. As much steam as possible is used in heating the feedwater to reduce the heat lost during start up. Any additional steam from the by-pass system is dumped to the main condensers. The ability to control feedwater temperature during start up is of value in controlling the heat pick up in the steam generator during start up and particularly when passing through the transition from water to steam.

The Breed unit is equipped with automatic bunker filiing equipment. This equipment conveys the coal from a surge pile in the storage yard to the two bunkers.

The control system is completely static using magnetic logic elements to perform the necessary control, interlocking, and protective functions. A bi-stable amplifier is used to control the coal feeder speeds from crusher motor current. This control circuit includes a time delay to compensate for the lag between a change in feeder speed and a change in crusher loading.

To date, there has been very little trouble with the logic elements. The belt speed measuring devices have been a problem particularly in freezing weather. The proximity switch used for level indication requires a paddle and linkage in the bunker. This linkage has jammed occasionally.

Before a fire can be lit in this once-through steam generator, a minimum flow of 875,000 lb per hour must be established. With this design, an additional requirement is that the natural circulation valve stem must be backed away from its disk after the minimum flow has been established. These interlocks incorporated with the conventional gas side interlocks provide the permissive firing interlock with boiler ignitors.

A tripping-sequence monitor is provided to aid in analyzing the causes of unit trip outs. This device provides a time sequence record of the operation of all tripping devices.

Steam temperature control during slag blower operation in the superheater is still under study. Interaction between the firing control and the attemperator controls occurs at this time and causes some cycling of the steam temperature. The most promising means of improvement appears to be revision of slag blower operating schedules to control the order in which temperatures change through the steam generator circuits.

Some trouble was experienced at the start in controlling flash tank level. Originally, the drain valves were sized to handle the full by-pass flow with virtually no pressure drop. As the steam generator is fired, these valves handle less and less flow with the pressure drop increasing to about 750 psi. A change was made to permit maintaining 175 psi air pressure in the flash tank until the by-pass flow became hot enough to flash into steam at this pressure. This permitted reducing the drain valve capacity appreciably with substantial improvement in the control response.

E. B. Morris then took on the subject of "Chemical Control and Cycle Clean-Up." The condensate clean-up system, also known as the condensate polishing system, was designed to operate in conjunction with the turbine by-pass system during start ups, and also during normal unit operation. It is arranged to carry out the following functions:

1) Clean or "polish" the condensate stream by filtration and demineralization.

2) Permit a deferred or orderly shutdown in the event of condenser leakage. Each condenser hotwell is divided into two sections for this purpose.

3) Polish evaporator vapor condensate for cycle make-up.

4) Clean contaminated heater and other drains which normally enter the cycle downstream of the filters.

5) Divert the entire condensate flow to storage at start ups, or divert up to one-third load condensate flow in normal operation. Contaminated and clean storage tanks were each sized at 500,000 gal for this purpose.

6) Prepare demineralized water from zeolite softened river water for the initial start up. No evaporator operation was possible until the unit was in operation.

The basic elements comprise four filters, operated in pairs, and three mixed bed demineralizers. These demineralizers operate at conventional flow rates for a total maximum of 675,000 lb per hour. This is equal to more than 60 per cent of start-up flow and about one-third of full load condensate flow. The demineralizers may not be operated except on pre-filtered water.

The filters employ wedge-wire wound tubular elements, each having 250 sq ft of surface area. Operating at a filtration rate of 4 gpm per sq ft, they permit 100 per cent filtration of full load condensate flow.

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Once again the Power Conference staged one of the more comprehensive meetings on the handling of condensate and on demineralization and flash evaporation.

The first of four sessions was opened by **Paul J. Hamm,** C. H. Wheeler Mfg. Co., with his paper "Effective Deaeration in Surface Condensers—Recent Experience."

The story of "A New Method for Increasing the Sensitivity of the Conductivity Measurement of Steam Purity" was presented by **R. W. Lane, C. H. Neff** and **T. E. Larson,** Illinois State Water Survey. Several recent publications, the authors announced, have suggested the inadequacy of the present conductivity methods for measurement of steam purity. This paper reports the application of basic principles to the measurement of conductivity and the excellent sensitivity and precision now obtainable for the determination of steam purity with these improvements.

Since the first application of the method, the conductivity of gases, normally present in steam and having no bearing on mineral carryover, has caused serious interference with the results. Ammonia and carbon dioxide are two of the gases known to cause serious interference with the conductivity measurement of steam condensate.

The authors then presented a series of figures showing the pH and conductivity relationships for mixtures of these gases. Computations for preparation of these figures were performed in the Digital Computer Laboratory of the University of Illinois. Since the equations have been formulated similar graphs, it is said, now can be easily prepared for other systems, such as the morpholine-carbon dioxide system, cyclohexylamine-carbon dioxide system and others, provided ionization constants and equivalent conductance values are available for the chemicals and temperatures in question.

Various devices, notably the Straub Degassing Condenser, have provided efficient degasification. However, with the advent of amine treatments for inhibition of corrosion, many of these devices have not eliminated amine interference. Since 1953, the Larson-Lane Steam Purity Analyzer, combining hydrogen exchange and degasification processes, has been used extensively to measure steam purity in the presence of amines. Essentially, this method consists of condensing steam at atmospheric boiling water temperature, of passing it through a hydrogen exchanger to remove amines and to convert boiler water salts to acids, and of recording the conductivity at boiling water temperature. While it is recognized that the conductivity of hydroxide and carbonate is lost by the ion exchange and reboil processes, the increased conductivity of other boiler water solids by conversion to the acid form materially increases the conductivity value in most cases. The method was originally designed to accurately record deviations from a normal recording of conductivity. Because the limits of impurities have not been completely or quantitatively defined, this method of measurement has proved satisfactory in many plants.

At higher steam pressures and temperatures, the purity requirements have accordingly become better defined, and the need for greater sensitivity and precision for determination of the absolute value for solids in the steam has become essential. J. K. Rice reported that sodium concentrations as low as 0.01 to 0.03 ppm may cause deposits in turbines and that the current conductivity techniques were insufficiently sensitive for such detection. These factors have stimulated the development of improved techniques for the recording and interpretation of conductivity measurements.

Using the data on the conductivity of the acids it is now possible to calculate the mechanical carryover in the steam from the specific conductance of the steam (in micromhos/cm at 98.5 C) and the boiler water anion concentrations.

It is recognized that this method of calculating an absolute value of dissolved solids in steam applies only to those cases in which the entire boiler water constituents in their same proportions become a part of the steam tested. Such conditions would be expected to occur when mechanical boiler water carryover occurs. Since the steam purity tests reported in this paper were conducted on steam from boilers of maximum pressure of 320 psig, vaporous carryover in which different constituents distill over in different proportions than in the original boiler water would not be expected. At higher pressures and temperatures evidence has been presented indicating that volatile carryover does occur. When data becomes available on the distilling characteristics of boiler water solids, the conductivity information on acids may then be used for calculating the dissolved solids of such steam.

M. E. Gilwood and **J. G. Mack,** The Permutit Company, presented the paper "Considerations Regarding Demineralization and Flash Evaporation of High Pressure Boiler Makeup Water." Demineralizing was adopted once a suitable process was developed, utilizing strongly basic anion resins that made it possible to produce economically a water having a low electrolyte and low dissolved silica content. The main advantage pointed out was the improved heat rate obtainable when employing demineralizing. The poorer heat rate obtained on the evaporator cycle was caused mainly by the "degradation" of the steam drawn from either the third or fourth extraction point of the turbine for passage through the evaporator. The loss in heat rate caused by this degradation of the steam in such cycles was presented in an earlier paper. The other major advantage of demineralizing over evaporators was that for the first time it made readily available to the power plant engineer a high quality water for the plant startup.

In the past few years the evaporator manufacturers have been endeavoring to eliminate the major disadvantages of evaporators in order to make them more acceptable for application in the regenerative cycle in high pressure condensing type turbine power plants. The flash evaporator with improved steam separation offers some promise for this purpose. To minimize degradation of steam in these evaporators the steam is drawn from the first extraction point of the turbine at low pressure. The flashed vapors are condensed by passing the feedwater through condensing coils in the evaporator, and the condensate is then passed into the main condenser.

The development of flash evaporators is an advance in the treatment of makeup water suitable for high pressure, low makeup, steam turbine plants. It minimizes one of the main disadvantages of submerged tube evaporators;

the considerable increase in heat rate encountered in submerged tube plants. Flash evaporators should have appreciable application especially over about 500 ppm.

Demineralizing appears more suitable on those waters having less than 500 ppm electrolytes. Demineralizing offers good treated water quality and ready availability of water for startups and cleanups. More experience data is awaited on flash evaporators.

(Editors note: A full length paper "Philadelphia Electric Co. Multistage Flash Evaporator" also presented at the Power Conference will be given in its entirety in the June issue.

Industrial Plant Savings

G. R. Chadwick, Consulting Engineer, with a paper on "Conservation of Utilities in the Small Boiler Plant" opened an interesting industrial plant session entitled "Economics of Power Plant Operation." Almost all of the plants would come close to 10,000 lb per hour, 200 psig or so.

Mr. Chadwick made the blanket statement "I dare say that there isn't a plant in existence where it isn't possible to save utilities. While in many instances the cost of doing the necessary work is too high in relation to the amount saved, there are many more cases, particularly in the small plants, where the saving possibilities are very high in relation to the investment required."

While emphasizing the importance of stopping small losses, the author felt it well to point out some common fallacies in what he considered the fuel saving "racket." One of the common fallacies is that smoke in itself represents an appreciable fuel waste. Actually, the per cent of loss in smoke is so small that it is never considered in computing a heat balance. Frequently, there is much more fuel going to waste from stacks that are as clear as the mountain air as there is from belching stacks.

In similar vein the author looked with critical eye upon the advertising messages for soot removal devices, water scale penalties. The inescapable moral of Mr. Chadwick's paper is that economy in engineering is a painstaking task requiring knowledge and much sweat.

R. H. Bradford, Ebasco Services, Inc., then took on the subject "Controlling the Cost of Electricity for Industrial Plants." If a plant man is purchasing electric power there are two things of primary interest: (1) is the billed rate, the proper one for the service, and (2) is the billed rate, the lowest possible through regulatory channels. As to the first, that is not simple; but there is a way to determine it—and at no cost to you. Call the power company industrial salesman responsible for your account and have him check the applicability of the rate.

The foregoing, Mr. Bradford emphasized, is really the least productive of all the ways to control your electric or power cost. What should be done is to check on: (1) reducing losses in the wiring system, (2) controlling billing demand by scheduling operations, either by preventing the building up of unnecessary peaks or by shifting some loads to "off peak," (3) checking loading on all motors and shifting motors to be sure they operate at near the point of best efficiency, (4) controlling and policing operations to ensure that motors, heaters, lights, etc., are off when not being used or needed, (5) when you are planning new equipment use electrical facilities.

The closing paper by **Myron B. Golber,** Armour and Co., gave helpful data on "Departmental Allocation of Utility Costs." "The usual approach for service departments within manufacturing or processing industries is to have their expenditures allocated by increasing the more obvious costs by a fixed percentage. Yet, the service department concept of distributing utility expense by increasing the more obvious costs by a percentagd fixed without regard to actual department or product utility requirements frequently leads to inequities· product prices and therefore product profits or losses have been affected," Mr. Golber stated.

Even where recognition has been given to distribution of utility expenses in relation to actual requirements, there is frequently another incorrect concept. This is that utility expenses rise and fall in direct proportion to increase or decrease of plant production. Ignoring seasonal effects, utility expenses do rise when production increases, and drop when plant production falls, but by no means in the same proportion.

When investigation has been made into the actual make up of the utility expenses, so that proof is obtained that they do not drop or rise in direct proportion to changes in plant production, the magnitude of the basic or non-variable portion of these costs has, in many cases, been startling. The fact that these expenses do not vary in proportion to production can especially be brought out in industries where air conditioning or refrigeration requires that humidities or temperatures be maintained in freezers, coolers, or other production areas even when processing is not taking place. Surveys at a number of Armour plants have shown basic expense to be from 40 to 60 per cent of total utility expense.

In starting allocation of utility costs, the point at which study begins is usually a recapitulation or report showing in itemized listing the expenses of all services involved. Recapitulations of utility expenses as supplied by accounting departments are frequently in the form of what might be called the ingredients of utility commodity expenses. That is, fuel expense is shown, but steam expense is not listed. Purchased electricity expense is lumped without regard to the department consumptions, nor are the accountants usually able to provide such breakdown. Water expense is shown for purchases, but the cost of pumping from wells or other local water supplies is hidden in electrical expense. Refrigerant expense is stated, but not the complete cost of refrigeration.

An average industrial plant may have as its utility commodities, steam (a separate cost usually required for each pressure level), electricity (purchased and generated costs separately, and then combined), cold and heated waters (usually separate costs for each temperature level, and frequently a separate cost for a non-potable supply for condensing or other special purpose), refrigeration (costs usually separated by system, or suction pressure level), and compressed air. Occasionally, hydrogen or inert gas generating plants are assigned to the power plant operators for handling, and then costs also must be supplied for these.

The first step is to allocate to each utility commodity the expenses, or portions of a combined expense item, with which is should be charged. A good place to start is with the incremental expenses; that is, those which reasonably may be expected to vary with plant production level.

Abstracts from the Technical Press—Abroad and Domestic

(Drawn from the monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)

Fuels: Sources, Properties and Preparation

A Correlation between Helium Density and Hydrogen Content of Coals. M. Mentser and S. Ergun. *Fuel* 1960, **39** (Nov.), 509–10.

Further investigations have confirmed that there is a linear relationship between the specific volume of coal and their atom per cent of hydrogen with natural graphite having zero hydrogen content. Both hydrogen content and specific volume can thus be used to establish the degree of coalification of high rank coals.

The Composition of the True Combustible Coal Substance. H. Romovackova and F. Kessler. *Erdöl und Kohle* 1960, **13** (Sept.), 639–46 (in German).

The different methods of conversion of coal to dry, ash-free substance are critically discussed. Studies of the changes of oxygen, carbon and hydrogen as a function of ash content are reported. It is concluded that it is simplest to carry out the conversion of the content of the individual elements of the same sample in two fractions of different ash content by a linear equation.

New Calorific Value Formulae for Bituminous Coal. W. Gumz. *B.W.K.* 1960, **12** (Nov.), 484–5 (in German).

Three new quadratic equations are proposed for calculating the gross and net C.V. of a bituminous coal. In the first the C.V. is a function of carbon content, in the second a function of volatile matter content and in the third a function of both C and M.V. The equations are believed to be more accurate than any existing equation.

Coke Combustion and Coke Reactivity. M. F. R. Mulcahy, A. Cameron and W. O. Stacey. *C.S.I.R.O. Coal Res. Rej. M.* **163** 1960 (July), 10 pp.

The influence of the porosity of a coke on its reactivity is discussed and experiments described in which the penetration of CO_2 into the pores was determined. It was found that this depends markedly on the reaction temperature. Preliminary results are presented.

Coal Storage. D. E. Shipley. *B.C.-U.R.A. Mon. Bull.* 1959, **23** (Oct./Nov.), 433–46.

This review deals with oxidation of coal in storage piles, observations on stock piles, laboratory tests of coal properties before and after storage, methods of storage and protection of stocks.

Hydraulic Transport of Solids in Pipelines. A. Redeker. *Fördern u. Heben* 1960, **10** (Oct.), 755–9 (in German).

After a brief review of existing pipelines in Russia, U.S.A. and Europe the equations developed by various authors for calculating the friction coefficient, water velocity, power requirements and specific gravity of the mixture are discussed for vertical and horizontal transport. Recommended values are tabulated. Means for entraining the material into the transport medium are described with special reference to the lock invented by Zänsler which appears to be the most economic and simplest apparatus available at present.

Heat Transmission with Boiling. G. S. Emmerson. *Nucl. Engng.* 1960, **5** (Nov.), 493–500.

The various theories of boiling heat transfer are presented, especially the transfer from nucleate to film boiling and burn-out. The application of the theory to pressurised and boiling water reactors is discussed.

Transient Natural Convection from Vertical Elements. B. Gebhart. *ASME Preprint* No. 60-HT-33 1960 (Aug.), 13 pp.

Equations are derived which give an accurate description of natural convection transients for vertical elements with time dependent internal energy generation.

Steam Generation and Power Production

A Survey of P-V-T- Data for Water in the Critical Region. E. S. Nowak, R. J. Grosh and P. E. Lilley. *ASME Preprint* No. 60-HT-25 1960 (Aug.), 13 pp.

A review of data for water from 2700 to 4000 psia, spec. volume from 0.025 to 0.125 cu ft/lb and temperatures from 630 to 800 F. There is a lack of smoothed or computed values for the spec. volume of superheated vapor and it is therefore difficult to derive accurate values of density and coefficients of expansion and compressibility. Data for superheated steam differ greatly.

Smoothed Pressure-Volume-Temperature Data for Water in the Critical Region Derived from Experimental Measurements. E. S. Nowak, R. J. Grosh, and P. E. Lilley. *ASME Preprint* No. 60-HT-26 1960 (Aug.), 13 pp.

Data derived from American, Russian and Czech measurements by a graphical method are presented for pressures from 3000 to 4000 psia, spec. volume of 0.030 to 0.12 cu ft/lb and temperatures from 690 to 752 F.

The Equation of State of Heavy Water on the Basis of Experimental P-V-T Data. A. M. Mamedov. *Teploenergetika* 1960, **7** (Sept.), 71–5 (in Russian).

The equation of state of heavy water is considered for heavy water in the range from 480 F to 700 F and 570 psig to 14,200 psig.

C.E.G.B. abstract.

The Viscosity of Steam and Water at Moderate Pressures and Temperatures. J. R. Moszynski. *ASME. Preprint* No. 60-HT-29 1960 (Aug.), 12 pp.

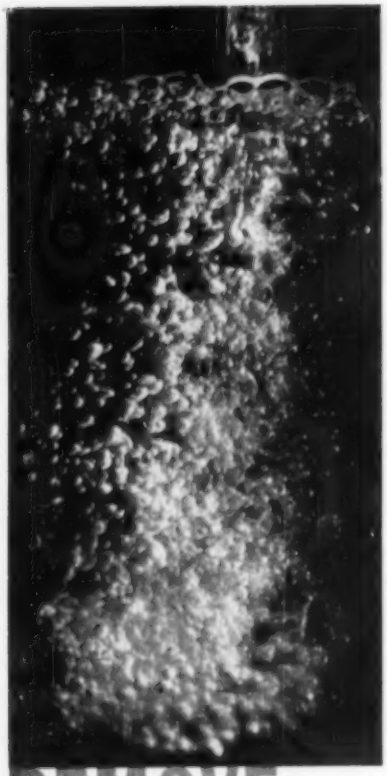
The viscosity of water has been measured from 3 to 340 atü and 20 to 186 C, that of steam from 1 to 42 atü and 137.5 to 270 C.

New Tables for the Specific Volume of Water to 1200 ata and 300 C. G. Sarukhanian. *B.W.K.* 1960, **12** (Nov.), 492–8 (in German).

The tables based on latest international research are those originally published by M. P. Voukalovich, B. H. Drampoff, D. S. Rasskasoff and S. A. Remisoff in Teploenergetika 1960 (July) 4–12.

Development Trends in German Steam Generator Design. H. Seidl. *V.G.B.* No. 68 1960 (Oct.), 285–96 (in German).

After a review of developments between 1920 and 1950 with the rapid increase of boiler size, steam pressure and temperature, introduction of pulverized fuel firing, of the forced-flow boiler and of slagging furnaces, boilers designed in recent years, all of forced-flow design, are described. A special feature is their prefabrication in the workshop; the advantages of this are outlined. The largest boilers installed at present for are 150 mw units, but those for 250 or 300 mw and larger are under consideration. The problem in Germany is the ability of operating during the night at 20–30 per cent of full load and for this reason two boilers to one turbine are often preferred. The application of temperatures above 100 F depends on technical developments and economics; only a few plants with steam temperatures of 100 F and above are



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operating. It appears possible that boilers for 1000 mw, 7000 psi and 1300 F with multiple reheat and incorporating a gas turbine cycle might be developed.

Development Trends in English Steam Generator Design since 1950. K. R. Lenel. *Mitt. V.G.B.* No. 68 1960 (Oct.), 297–310 (in German).

The position of the electricity supply industry at the time of nationalisation is outlined and the reasons for the rapid increase of unit and power station size in the years 1950–1960 presented. Designs of boilers from 180 to 3750 klb/h, mills, burners, casings and ash handling plant are described. The introduction of forced-flow boilers, especially those operating at supercritical pressure, is discussed. Future developments are difficult to forecast in view of the uncertain trends of nuclear power generation.

Development Trends in the Design of Steam Generator Plants of the French Public Energy Supply. Y. Teste. *Mitt. V.G.B.* No. 68 1960 (Oct.), 310–21 (in German).

The position of public steam power stations, hydro-electric generation and power plants of the coal mining, iron and steel and other works supplying the public network is outlined. Examples of standardised boilers rated at 800–885 klb/hr, 1800 psi and 1004/1004 F for 125 mw units and those rated at 1550–1700 klb/h, 2760 psi and 1050/1050 F for 250 mw units are described. The fuel is mostly coal, but oil and natural gas are increasingly used. Details of designs of controls, air preheaters (tubular), finned tubes for superheaters and reheaters prefabrication and multi-tube heat exchangers for nuclear reactors are presented.

Development of the Benson Boiler in Germany. R. Mutke. *Heat Engng.* 1960, **35** (Sept./Oct.), 168–73.

Examples of Benson boilers designed by VKW from its first inception to the present time are presented.

The TPP-110, a New High-output Boiler. V. S. Patychenko, I. N. Goldenfarb and L. A. Ostrovskii. *Energomashinostroenie* 1960, **6** (Aug.), 1–11 (in Russian).

Full details are given of the boiler which has an output of 935 t/h at 3750 psig and 1080 F. The plant is built up as a series of self-contained units of exceptional size.
C.E.B.G. Digest 1960, **12** (Dec. 3), 3169.

Boiler Blowdown. W. F. Gerrard. *Pwr. Wrks. Engng.* 1960, **55** (Nov.), 870–6.

The reasons for the necessity of blowing down boilers are explained. The various methods used for this and the determination of the amount to be blown down are described and illustrated by an example.

Liquid and Gaseous Fuel Firing

The Economic Combustion of Sulfur Containing Fuel Oil, a Means of Avoiding Dew Point Difficulties in Boiler Operation. F. Glaubitz. *Mitt. V.G.B.* No. 68 1960 (Oct.), 338–43 (in German).

Severe difficulties in the operation of boilers fired by heavy fuel oil with high S content could be overcome by redesign of the burners which resulted in rapid ignition and complete combustion with minimum excess air. Part of the air is introduced into the burner as core air, the larger part at high velocity parallel to the oil jet and without swirl to prevent the air from flowing away from the flame. The CO_2 content of the flue gas averages 15.3 per cent and the O_2 content 0.2–0.4 per cent. It has not been found necessary to use the soot blowers and steel shot cleaners; all heating surfaces including the tubular air preheater have remained clean.

Water-Side Corrosion and Water Treatment

Present-Day Feed-Water Treatment for High-Pressure Boilers. P. Hamer. *Proc. Inst. Mech. Engrs.* 1960, **174** No. 1, 69–94.

The quality of feedwater required by industrial boilers operating at 1000–2000 psi is discussed and present treatment methods described. The control of steam purity, concentration of salts and silica in the boiler is dealt with. Corrosion in high-pressure boilers and preventative measures are discussed, especially for boilers standing during erection and normal operation.

Possibilities of Application of Filter Aids in Water Treatment. H. Hannig. *Mitt. V.G.B.* No. 68 1960, (Oct.), 332–8 (in German).

The various filter aids, filter installations and their operation are discussed. Specially developed plants for mechanical filtration and removal of iron from water, removal of oil from condensate, reduction of organic substances from water by filter aids are described. Control of the filtrate is obtained by measuring the turbidity with an instrument based on the Tyndall effect.

Corrosion of Boiler Tubes Subjected to Temperature Changes. D. Ya. Kagan and L. S. Zhuravlev. *Teploenergetika* 1960 (Feb.), 60–7, *D.S.I.R. Transl.* RTS 1602.

Corrosion of tubes in the transition

zone from water to steam in Sulzer monotube boilers was found to be due to the destruction of the oxide film by sharp variations of wall temperatures during phase changes. Similar observations were made in Russian forced flow boilers.

Gas-Side Corrosion and Deposits

An Aspect of the Corrosion of Mild Steel by Combustion Gases. T. K. Ross, A. J. Macnab and B. E. Leyland. *J. Inst. Fuel* 1960, **33** (Nov.), 540–2.

The corrosion of mild steel by SO_2 in a reducing atmosphere above dew point with pure N_2 and added H_2 and O_2 has been studied. The scaling rates are expressed as a function of sulfur concentration and temperature.

Electro Precipitators: Practical Design Aspects. J. E. Sayers. *J. Inst. Fuel* 1960, **33** (Nov.), 542–50.

Precipitator performance factor, factors outside and under designer's control and designs for power stations and steel works are considered.

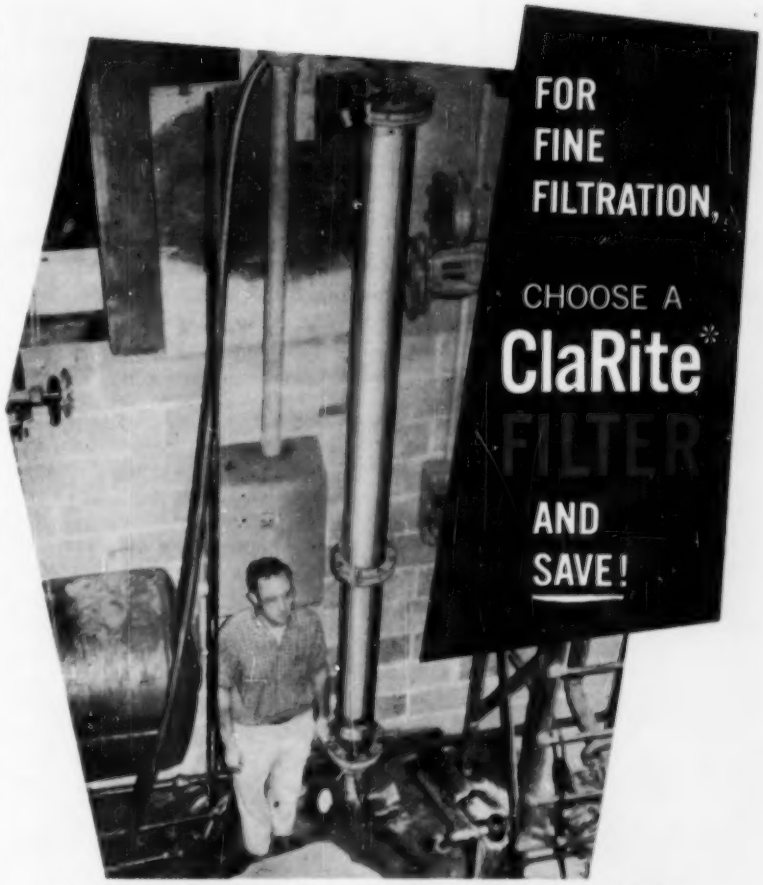
Considerations on the Usefulness of Application of Electrostatic Precipitators and Mechanical Fly Ash Separators Upstream or Downstream for the Cleaning of Flue Gases. R. F. Heinrich. *Mitt. V.G.B.* No. 68 1960, (Oct.), 322–32 (in German).

The changes in the composition of the fly ash over the years is traced and its effect on the separation efficiency of electrostatic precipitators based on studies carried out by Simon-Carves Ltd. described. Improvement of the separation efficiency by the installation of cyclone separators upstream or downstream of the precipitator is discussed; the decision which combination to employ will depend on the properties of the fly ash and local conditions. In many cases especially in new plants, better results will be obtained by using a multi-stage precipitator than a simple precipitator and cyclone combination.

Note: Experience with Shot-cleaning Plant for Air Preheaters. A. C. Kjaer. *J. Inst. Fuel* 1960, **33** (Nov.), 553–4.

Attempts were made in stoker-fired boilers with additional oil firing to overcome tubular air preheater clogging by either steam soot blowing or shot cleaning. The former increased smut emission and was abandoned, the latter with 3 × 2 h operation in 24 h kept the heater clean when firing up to 10 per cent of oil on a Btu basis, but above this the shot became covered in sulfurous soot.

Note: Experience in the Cleaning of Economizers on Power Station Boilers by the Shot-cleaning Method. F. W. Board and J. P. Holmes. *J.*



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4. two direct-to-drum assemblies, one on each end of the drum.
5. use one conventional gage, either direct-to-drum or on water column, and two manometric-type remote gages (the Reliance EYE-HYE) if operating over 900 psi. (Case No. 1155.)

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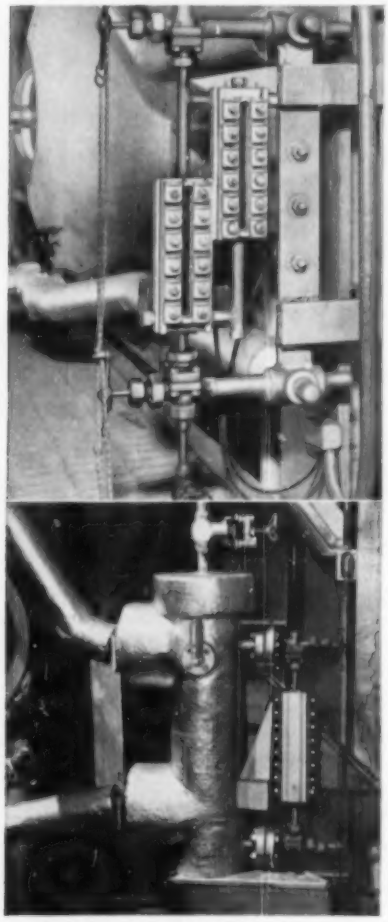

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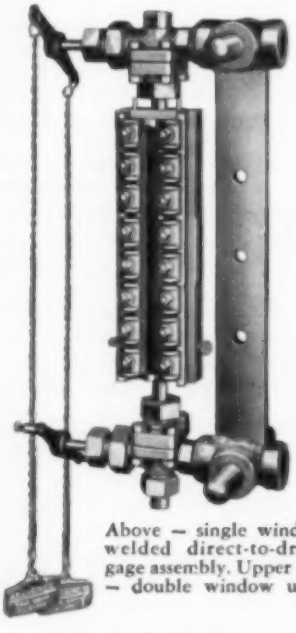

Above — single window welded direct-to-drum gage assembly. Upper left — double window unit.

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Inst. Fuel 1960, **33** (Nov.), 554–8.

The shot-cleaning system and full automatic control gear at Battersea power station are described. It is estimated that the regular on-load cleaning of the economizer will lead to a fuel saving of 1.1 per cent.

Further Corrosion Tests on Materials in Regenerative Air Preheaters. J. F. Barkley, H. Karlsson and C. F. Stark. *U. S. Bureau of Mines.* Report No. 5471, 1959, 28 pp.

Field tests were made on the rate of corrosion of 30 materials used in air preheaters under regular operating conditions, and the buildup of deposits was observed. Twenty-three materials were tested with coal as the fuel on an underfeed stoker and seven with oil, including a short period with pulverized coal as fuel. The plant equipment is described, and pertinent plant operating data are tabulated.

From C.E.B.G. Digest 1960, **12** (Nov. 19), 3156.

Power Generation and Power Plant

Central Station Design Survey. Anon. *Power* 1960, **104** (Oct.), 161–72.

The annual review of units planned, under construction and starting operation. It is now predicted that capacity will grow at a rate of 5 per cent annually.

Industrial Plant Design Survey. Anon. *Power* 1960, **104** (Oct.), 89–108.

The annual review covers electrical systems, steam conditions, fuels and firing appliances, boiler and turbine sizes, controls, subdivided into plants over and below 150 psi. Combination gas and oil firing is still preferred, whilst coal firing by underfeed and spreader stoker declines. More small than larger boiler sizes are being installed and electric combustion controls are on the increase.

The Supercritical Power Station Unit Asserts Itself. K. Schröder. *B.W.K.* 1960, **12** (Nov.), 499–500 (in German).

A brief illustrated description of the unit in Breed power station originally rated at 450 mw and now uprated to 475 mw is given. The net heat rate is 8500 Btu/kwh, the thermal efficiency 40 per cent and the volume 12 cu ft³/kw. The orders for supercritical pressure units at Drakelow 'C' and Russian intentions of installing 28 300 mw units at 3356 psi and 1076/1050 F are discussed and American, British and German units in operation or on order tabulated giving output, pressure, temperature, turbine output and boiler supplier.

Aberthaw Power Station. W. Wilde. *Proc. Sth Wales Inst. Engrs.* 1960, **75** (Oct.), 117–32.

The Aberthaw power station is designed for a capacity of 600 mw consisting of six 100 mw turbogenerator units. The single drum, natural circulation 755 klb/hr boilers are rated at 1600 psig and 985 F reheat. The circulating water system, p.f. firing system, dust and ash handling plants, turbines, generators, feed water system, and electrical plant are described. The economy in space and cost effected by building larger units is indicated by tabular data and graphs.

From C.E.B.G. Digest 1960, **12** (Nov. 19), 3026.

Korneuburg Power Station. Anon. *Engineer* 1960, **210** (Nov. 11), 815–6.

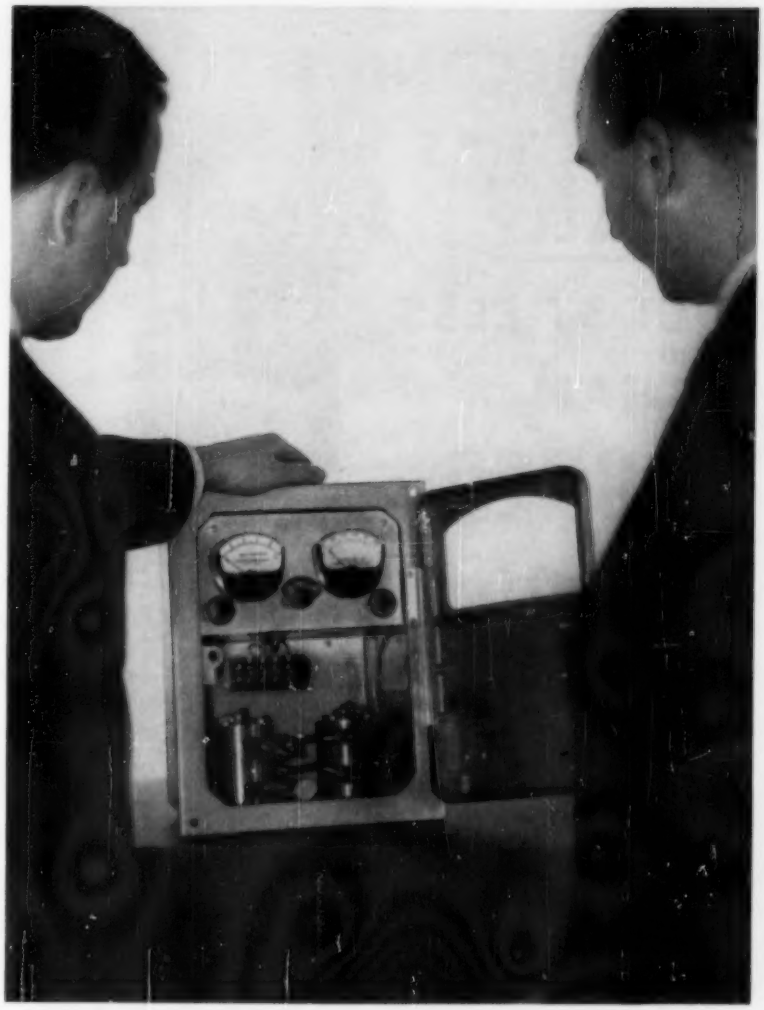
This power station on the banks of the Danube in Austria contains two 25 mw double-shaft gas turbines, fired by natural gas; two waste-heat boilers fired by hot exhaust gases and additionally by oil or gas each supply 110 klb/h at 213 psi and 842 F to a common 25 mw turbogenerator. There is further a Benson boiler rated at 530 klb/h at 1877 psi and 995/986 F to supply a 77 mw turbogenerator; this boiler is also oil or gas fired.

A Coal-Fired Thermal-Storage Boilerhouse. Anon. *Pwr. Wrks- Engng.* 1960, **55** (Nov.), 880–9.

At the British Oil and Cake Mills Ltd.'s Olympia mills the highly fluctuating steam demand is met by four travelling grate stoker fired shell-type boilers each rated at 17,000 lb/h at 175 psi and 530 F, the water space being large enough to allow a fall or rise in the water level of 2–3 ft. The controls keep the pressure within ±1 psi. Sootblowers are installed in the furnace, in front of the smoke box and behind the bridge wall of the stoker. The flue gas are cleaned by a suction type grit arrestor and a secondary collector. The boiler ash is removed by ash extractor under each stoker into a hydraulic system. The boilers are continuously blown down into a flash tank, the steam being used to heat the feed water and condition the coal. The water is treated by flocculation, settling and filtration followed by an ion starvation, base exchange plant.

Developments in the U.S.S.R. Anon. *Weld. Met. Fabr.* 1960, **28** (Nov.), 457–64.

Impressions of the welding art as practiced today in Russia are given following a visit of a team of British experts. Of special interest have been the progress in electroslag welding and automatic submerged arc welding.



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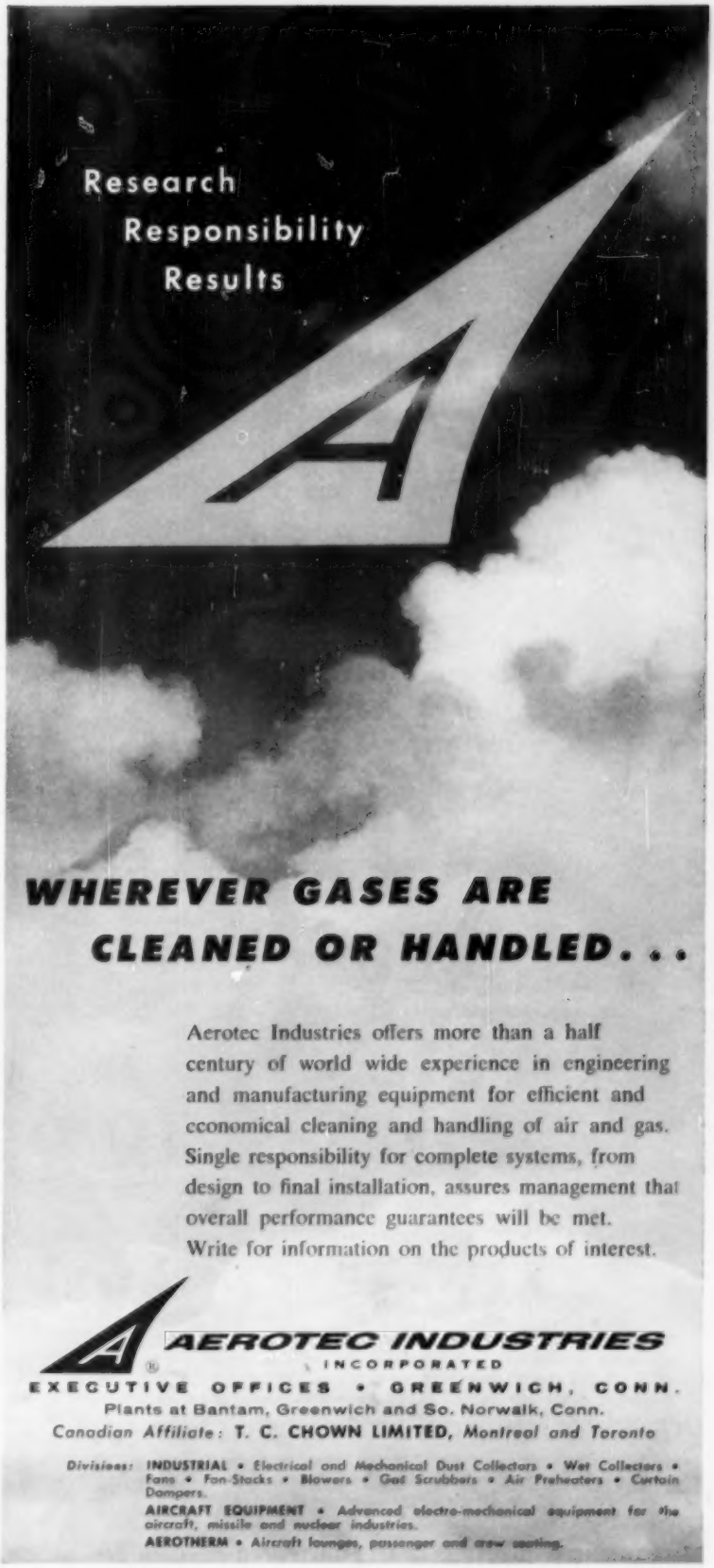
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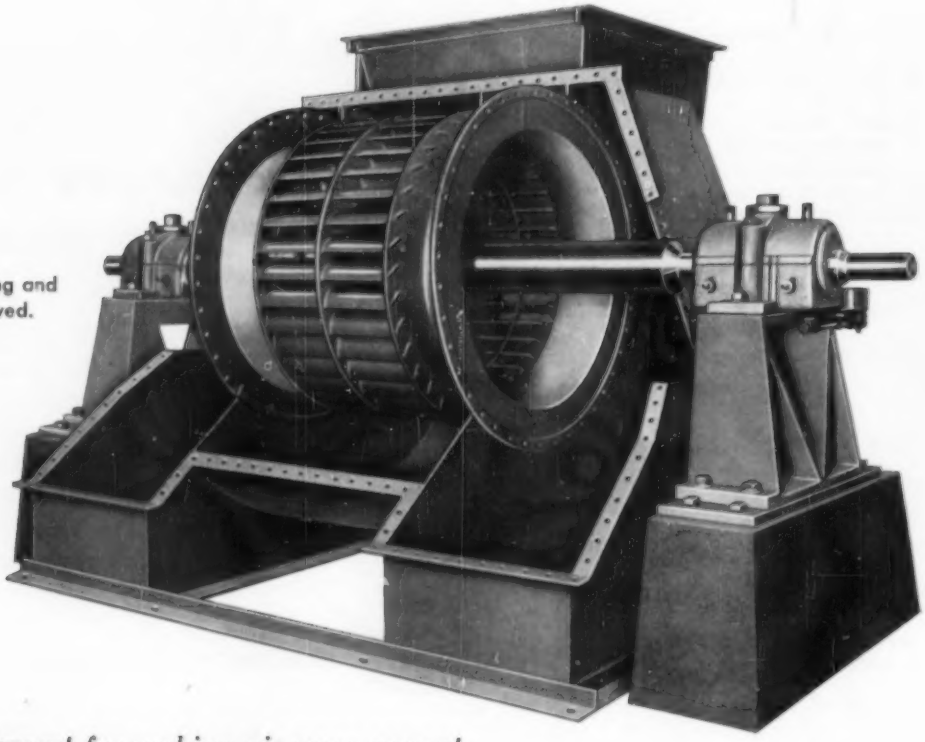

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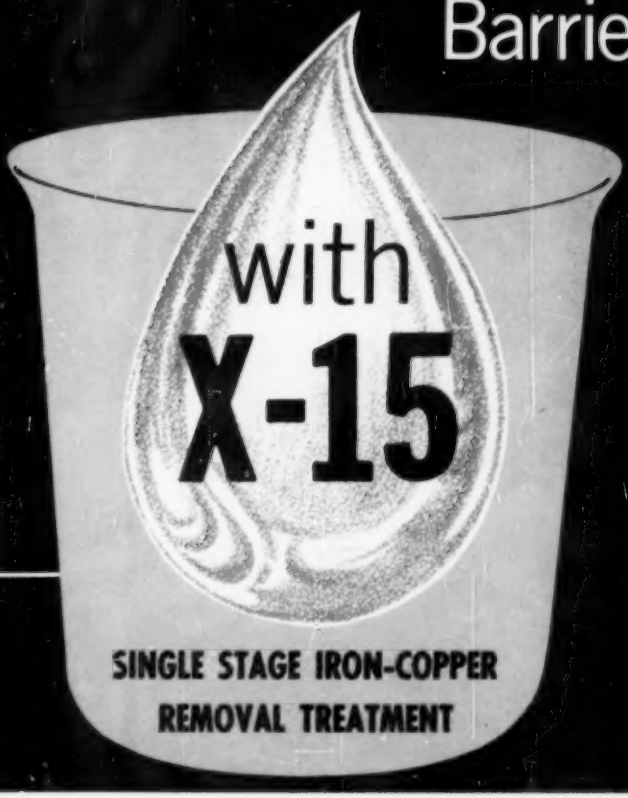

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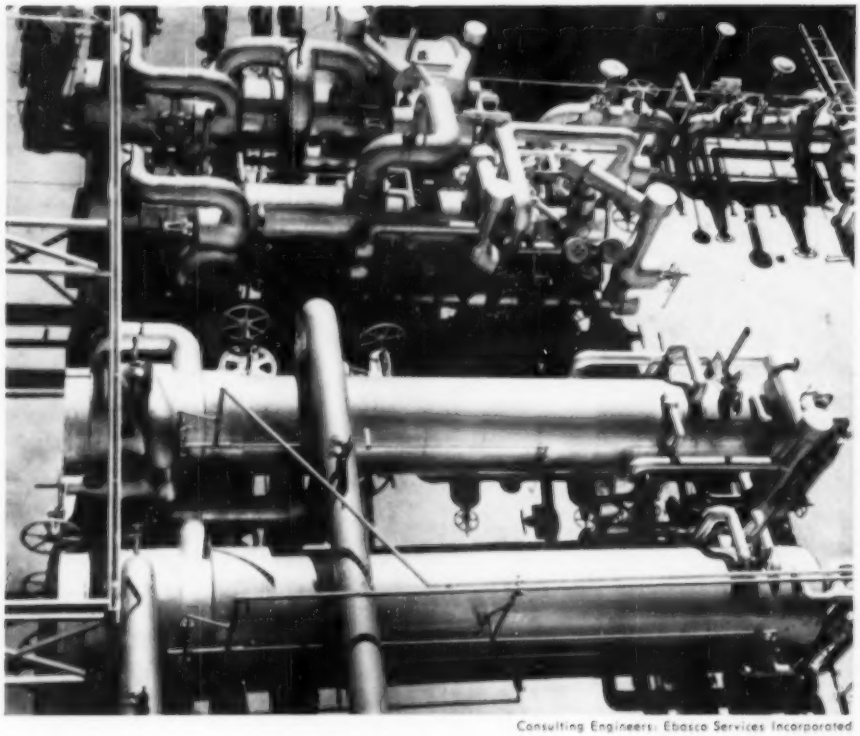

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